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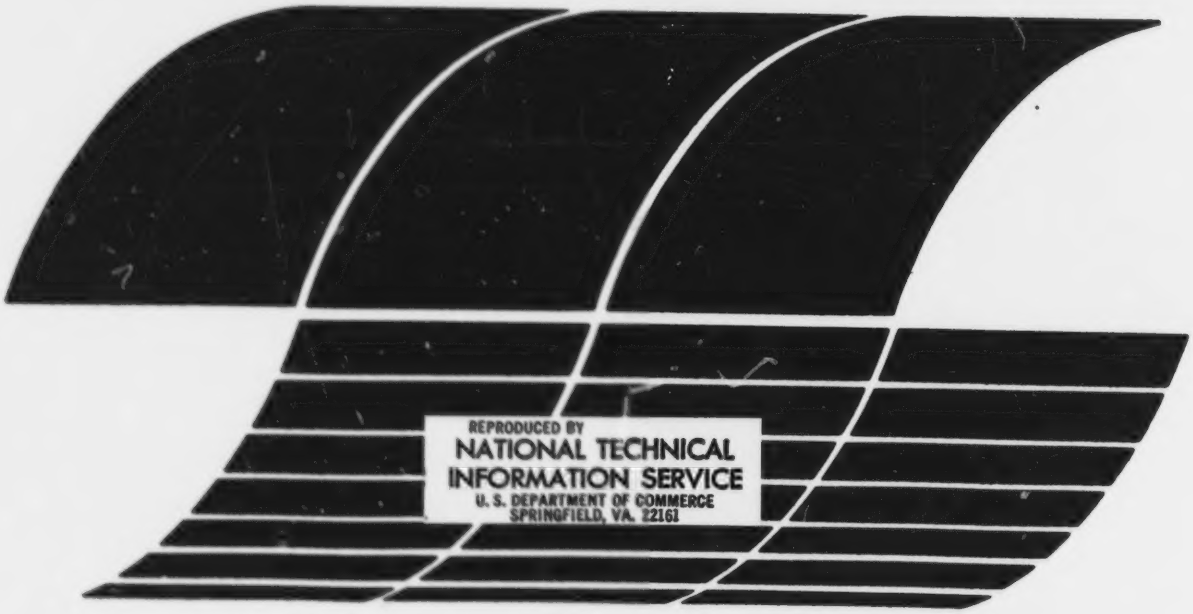
Volume III.
Stationary Engine and Industrial Process Combustion Systems

Interagency Energy/Environment R&D Program Report



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16. ABSTRACT The proceedings document the approximately 50 presentations made during the symposium, March 5-8, 1979, in San Francisco. Sponsored by the Combustion Research Branch of EPA's Industrial Environmental Research Laboratory-RTP, the symposium dealt with subjects relating both to developing improved combustion technology for the reduction of air pollutant emissions from stationary sources, and to improving equipment efficiency. The symposium was in seven parts, and the proceedings are in five volumes: I. Utility, Industrial, Commercial, and Residential Systems; II. Advanced Processes and Special Topics; III. Stationary Engine and Industrial Process Combustion Systems; IV. Fundamental Combustion Research and Environmental Assessment; and V. Addendum. The symposium provided contractor, industrial, and government representatives with the latest information on EPA inhouse and contract combustion research projects relating to pollution control, with emphasis on reducing NOx while controlling other emissions and improving efficiency.

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Combustion	Nitrogen Oxides	Stationary Sources	21B	07B
Field Tests	Efficiency	Environmental Assessment	14B	
Assessments	Utilities			
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and
Robert E. Hall, Symposium Vice-chairman

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PREFACE

These proceedings document more than 50 presentations and discussions presented at the Third Symposium on Stationary Source Combustion held March 5-8, 1979 at the Sheraton Palace Hotel, San Francisco, California. Sponsored by the Combustion Research Branch of the EPA's Industrial Environmental Research Laboratory - Research Triangle Park, the symposium papers emphasized recent results in the area of combustion modification for NO_x control. In addition, selected papers were also solicited on alternative methods for NO_x control, on environmental assessment, and on the impact of NO_x control on other pollutants.

Dr. Joshua S. Bowen, Chief, Combustion Research Branch, was Symposium Chairman; Robert E. Hall, Combustion Research Branch, was Symposium Vice-Chairman and Project Officer. The welcoming address was delivered by Clyde B. Eller, Director, Enforcement Division, U.S. EPA, Region IX and the opening Address was delivered by Dr. Norbert A. Jaworski, Deputy Director of IERL-RTP.

The symposium consisted of seven sessions:

Session I:	Small Industrial, Commercial and Residential Systems Robert E. Hall, Session Chairman
Session II:	Utilities and Large Industrial Boilers David G. Lachapelle, Session Chairman
Session III:	Advanced Processes G. Blair Martin, Session Chairman
Session IV:	Special Topics Joshua S. Bowen, Session Chairman
Session V:	Stationary Engines and Industrial Process Combustion Systems John H. Wasser, Session Chairman
Session VI:	Fundamental Combustion Research W. Steven Lanier, Session Chairman
Session VII:	Environmental Assessment Wade H. Ponder, Session Chairman

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SESSION V

STATIONARY ENGINES AND INDUSTRIAL PROCESS COMBUSTION SYSTEMS

JOHN H. WASSER
SESSION CHAIRMAN

APPLICATION OF ADVANCED COMBUSTION MODIFICATIONS TO INDUSTRIAL PROCESS EQUIPMENT
--PROCESS HEATER SUBSCALE TESTS

By:

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ABSTRACT

This is a report of a research program to develop combustion modification technology as means of emissions reduction and thermal efficiency improvement on industrial process equipment. The work is an extension to EPA Contract 68-02-2144 which concentrated on operational adjustments. Presented are results of subscale tests for petroleum refinery heaters.

Subscale tests with natural draft process heater burners firing natural gas, No. 2 oil, No. 6 oil and shale oil included standard burners, two commercial low NO_x designs, staged combustion, flue gas recirculation, steam injection, and altered fuel injection. The most effective of these for reducing NO_x was staged combustion; reduction obtained with natural gas was 67% from a baseline of 61 ng/J (120 ppm, 3% O_2, dry) and with No. 6 oil (0.3% N_2) was 51% from a baseline of 172 ng/J (307 ppm, 3% O_2, dry). The costs of applying all modifications were evaluated; stage combustion appears to be the most cost effective for large heaters while commercial low-NO_x burners are more cost effective for small heaters burning No. 6 oil.

These subscale test results are preparatory to full scale testing and should not be interpreted as achievable technology until full scale test demonstration is completed.

This report was submitted in fulfillment on Contract 68-02-2645 by KVB, Inc. under the sponsorship of the U.S. Environmental Protection Agency. This report covers the period August 5, 1977 to October 30, 1978.

SECTION 1

INTRODUCTION

At the Second EPA Stationary Source Combustion Symposium, KVB reported on work to investigate the feasibility of applying combustion modifications, developed for boilers, to other industrial process equipment (Ref. 4). Results of that program were mixed. Some units responded well with up to 69% reduction in NO_x. For other types, equipment design and process constraints limited NO_x reduction to insignificant or negligible amounts. This paper presents additional work directed to understand and overcome these limitations.

OBJECTIVE AND SCOPE

The objective of this program is to develop advanced combustion modification concepts requiring minor hardware modifications that could be used by operators and/or manufacturers of selected industrial process equipment to control emissions. The development is to be performed for equipment in which the modifications will be most widely applicable and of the most significance in mitigating the impact of stationary source emissions on the environment. The program involves investigation not only of emissions but also multimedia impacts and control cost effectiveness.

The program involves both subscale and full scale testing. Subscale testing is a necessary part of development of new hardware to ensure acceptable performance which is a vital aspect of emissions control. Full scale testing is also necessary on more than one process design configuration (e.g., forced draft and natural draft) before the equipment manufacturers and process industry can employ an emission control technology.

At the conclusion of the study, a final engineering report will be prepared summarizing the accomplishments of the subscale and full scale demonstration tests. A series of guideline manuals will be prepared to acquaint the equipment manufacturers with the most promising emission control methods that have been demonstrated and to offer technical guidance that can be directly applied in their process equipment design.

PRELIMINARY SURVEY

The initial task was to review existing source inventories and update them where possible to more clearly define those processes where controls will be of maximum benefit. Activity in the preliminary survey task concentrated on the review of processes to be emphasized in the test program. The review of source emission NO_x data and total annual heat input provided the following relative ranking of industrial processes as most important in potential environmental impact:

Process	NO_x Emissions Gg/y	NO_x Emissions 10^3 tons/y	Heat Input 10^{15} J/y	Heat Input 10^{12} Btu/y
1. Cement kilns	704	776	513	486
2. Wood-bark boilers	141	157	915	873
3. Refinery process heaters	122	134	1580	1500
4. Glass container furnaces	39	43	105	99
5. Steel soaking pits and reheat furnaces	29	32	538	510

Table I presents additional key information used to develop comparative data on each process.

Of the five processes considered, four were identified as candidates for combustion modification. Glass furnaces were excluded because of a lack of flexibility in the combustion systems.

TEST PROGRAM STATUS

Subscale test work has now been completed on a vertically-fired rectangular process heater research furnace and a small research dry process rotary cement kiln. The process heater subscale tests were conducted in conjunction with a major process heater burner manufacturer. The research kiln tests were conducted with the help of a major cement industry association.

Instrumentation used is discussed in Section 2.

In chronological order, the modifications tested in the subscale process heater were the following:

1. Lowered excess air
2. Commercial low-NO_x burners (two designs)
3. Steam injection
4. Staged combustion (two methods)
5. Flue gas recirculation
6. Modified fuel injection

Each of these concepts is defined in Section 3 of this paper. A summary of test results for each modification is also given in that section. An analysis of the cost effectiveness of each modification is presented in Section 4.

A summary of the results of the tests on the process heater is presented in Table II. This table shows that relatively minor hardware modifications may be quite effective in reducing NO_x in process heaters (e.g. reductions in NO_x of 50-60% or more were obtained using fairly simple staging techniques as compared to a maximum 63% reduction for the more complicated flue gas recirculation technique).

Tests on the research kiln are not expected to be representative of producing cement kilns, however, preliminary indications are that fly ash injection into the flame zone is effective in reducing NO_x. Reductions of 28% in NO_x emissions were achieved with fly ash injection. A basic objective in these tests was to evaluate the effect of the modifications on cement quality. Analyses of cement samples were not yet available for discussion in this paper. Therefore, the results of those tests will be presented in a later paper and the final report.

SECTION 2

INSTRUMENTATION

The process heater emission measurements were made with instrumentation carried in a 32 ft x 8 ft mobile laboratory which was described in detail in the EPA Report, "Application of Combustion Modification to Industrial Combustion Equipment," Contract No. 68-02-2144.

Gaseous species measurements were made with analyzers located in the trailer. Particulate emission and size measurements were not made during subscale tests to allow larger range of test variables for effect on gaseous emissions. These measurements will be made on full scale units. The emission measurement instrumentation used is the following:

TABLE III. EMISSION MEASUREMENT INSTRUMENTATION

Species	Manufacturer	Measurement Method	Model No.
Hydrocarbon	Beckman Instruments	Flame Ionization	402
Carbon Monoxide	Beckman Instruments	IR Spectrometer	865
Oxygen	Teledyne	Polarographic	326A
Carbon Dioxide	Beckman Instruments	IR Spectrometer	864
Nitrogen Oxides	Thermo Electron Co.	Chemiluminescent	10A
Sulfur Dioxide	DuPont Instruments	UV Spectrometer	400
Smoke Spot	Bacharach	ASTM D2156-65	21-7006

GAS SAMPLING AND CONDITIONING SYSTEM

The flue gas sampling system uses positive displacement diaphragm pumps to continuously draw flue gas from the stack into the laboratory. The probes are connected to the sample pumps with 0.95 cm (3/8 in.) or 0.64 cm (1/4 in.)

nylon line. The positive displacement diaphragm sample pumps provide unheated sample gas to the refrigerated condenser (to reduce the dew point to 35 °F), a rotameter with flow control valve, and to the O_2, NO, CO, and CO_2 instrumentation. Flow to the individual analyzers is measured and controlled with rotameters and flow control valves. Excess sample is vented to the atmosphere.

To obtain a representative sample for the analysis of NO_2, SO_2 and hydrocarbons, the sample must be kept above its dew point, since heavy hydrocarbons may be condensible and SO_2 and NO_2 are quite soluble in water. For this reason, a separate, electrically-heated, sample line is used to bring the sample into the laboratory for analysis. The sample line is 0.64 cm (1/4 in.) Teflon line, electrically traced and thermally insulated to maintain a sample temperature of up to 400 °F. Metal bellows pumps provide sample to the hydrocarbon, SO_2 and NO_x continuous analyzers.

SECTION 3

COMBUSTION MODIFICATIONS ON A SUBSCALE PROCESS HEATER

EQUIPMENT CHARACTERISTICS

The process heater subscale testing was conducted in the research furnace of a major manufacturer of process heater burners. The furnace was a refractory lined, uncooled rectangular box type furnace 2.4 m (8 ft) wide by 1.8 m (6 ft) deep by 9.8 m (32 ft) high.

Natural draft burners were installed in the furnace floor firing vertically upward. The nominal firing rate for the tests was 1.5 MW ($5\text{x}10^6$ Btu/hr). Furnace draft was controlled manually with a damper in the stack. View ports for observing flame shape were provided.

The furnace had the capability of firing either oil or natural gas. Both flows were measured with flow meters. Thermocouples were installed in the side of the furnace to measure the vertical thermal gradient and the temperature-time behavior during furnace heat-up.

BURNER COLD FLOW TESTS

A cold flow burner model with the same horizontal dimensions as the natural draft heater was fabricated at the KVB laboratory. The purpose of the cold flow tests was to evaluate the fuel and air mixing to provide insight into fuel injection modifications which could lead to lowered NO_x emissions when firing natural gas.

The cold flow model used an induced draft fan to obtain the same air flow and velocity as that of the actual natural draft burner. The natural gas was simulated by gaseous CO_2. Measurements of CO_2 concentration were made in two axes across the firebox at three axial planes. Contour maps of constant concentration were prepared to compare the mixing characteristics of the different gas tips. After the mixing model was prepared, modifications to the gas tips were made and tested. The gas fuel was simulated by CO_2 and the concentration measured with an NDIR CO_2 analyzer. Actual burner fuel/air ratio and the cold flow simulation are realted by the following expression:

$$(F/A)_{\text{Burner}} = (F/A)_{\text{Sim.}} \left[(\rho_f/\rho_{air})_{\text{Burner}} \, (\rho_{air}/\rho_f)_{\text{Sim.}}\right]^{1/2}$$

Tests were conducted with three types of standard gas tips, each providing a different degree of swirl, supplied by the burner manufacturer. Very little difference in mixing patterns was found among the three standard types of gas tips. All produced substantial mixing of the fuel gas and air within a short distance of the injection plane.

Several modifications to the fuel injection geometry were evaluated. Modifications which looked promising from the cold flow model were then evaluated in the hot firing tests in the furnace.

The modifications tested were the following:

1. Turning the gas tips so that the center gas port was aimed radially outward such that the gas stream impinged upon the 41 cm (16 in.) diameter cylindrical sleeve.
2. Placing a 20 cm (8 in.) diameter 'staging' cylinder with its vertical centerline coinciding with that of the burner into the flow such that roughly 25%-30% of the 'combustion' air flow was introduced through the cylinder. Two cylinders of different length were used in separate tests. In one case, the top of the cylinder was 5.4 cm (2-1/8 in.) above the gas tips. In the second case, the cylinder top was 30.5 cm (12 in.) above the gas tips.
3. Placing a 7.6 cm (3 in.) wide, 15.2 cm (6 in.) long deflector upstream of each of the gas nozzles inclined at a 45 degree angle from vertical and extending from the 'burner' sleeve to the plane of the gas tip orifices.

All of these modifications were expected to delay mixing of fuel and air, thereby lengthening the flame in a hot-firing application, lowering peak temperatures and residence times at high temperatures and, thus, lowering NO_x emissions. The concentration of the cold flow test gas (CO_2) was again measured at various radial positions at three heights above the gas injection plane. The results for the four different test cases at 36 cm (14 in.) above the gas nozzles are shown in Figure 1. These curves indicate that the concentric 'staging' cylinder and the radially-outward-facing injection orifices produce a significant delay in the mixing of the test gas and air. The mixing pattern with the deflectors in place did not vary appreciably from the patterns obtained for the standard nozzles without modification.

HOT FIRING TEST RESULTS

Tests were conducted to evaluate the effect of combustion modifications on emissions from a natural draft process heater. The reduction in NO_x emissions and the change in efficiency were evaluated for: (1) lowered excess air, (2) staged combustion air, (3) low NO_x burners (tertiary air injection and recirculating tile designs), (4) flue gas recirculation, (5) steam injection and (6) altered fuel injection geometry. The tests were conducted with natural gas and No. 6 oil. Only burner baseline measurements were made with No. 2 oil because of limited furnace availability due to the manufacturer's work schedule.

Baseline Tests

Tests were conducted with each burner prior to implementing any combustion modification. These baseline measurements were made with the burner firing natural gas, No. 6 oil (0.3% N) and No. 2 oil (0.01% N). A summary of baseline gaseous emissions data is presented in Table IV.

Lowered Excess Air (LEA)

The NO_x emissions from the conventional MA-16 and DBA-16 burners as well as the low-NO_x (tertiary air) burner for various test conditions including baseline are graphed as a function of stack excess oxygen for natural gas and No. 5 oil firing in Figures 2 and 3. The unmodified burners are represented by heavy curves and the results of the modifications are shown as light curves. The low NO_x burner (tertiary air injection) exhibited the lowest level of NO_x at the nominal 3% O_2 condition and showed the most dependence on O_2 level firing natural gas. The NO_x emissions at 2.7% O_2 was 100 ppm* which dropped off to 76 ppm at 2.1% O_2.

The effect of excess O_2 on NO_x for No. 6 oil firing with a 40 degree spray angle is shown in Figure 3 for the MA-16 and the tertiary air natural draft burners. The spray angle is defined as the total included angle of the conical jet produced by the oil gun. The minimum value for NO_x for the MA-16 was 282 ppm at 0.75% O_2. The minimum NO_x emission for the tertiary air burner firing No. 6 oil was 235 ppm at 0.5% O_2.

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*All concentrations in this report are corrected to 3% excess O_2 on a dry basis.

Staged Combustion Air (SCA)

Staged combustion is a technique for emissions reduction wherein a portion of the flame zone is operated fuel-rich and secondary air is injected subsequently to bring the overall air-fuel ratio to the desired level to assure complete combustion. Staged combustion has been shown to be an effective method of NO_x reduction in other applications. In order to develop staged combustion in a natural draft heater, two techniques were evaluated. In the first method, four staged air lances were inserted through the furnace floor positioned 90 degrees apart outside the burner tile on a diameter of 61 cm (24 in.). This modification is shown schematically in Figure 4. The staged air lances were made from 3.2 cm (1-1/4 in.) diameter stainless steel pipe with an orifice plate of 3.0 cm (1-3/16 in.) diameter on the end. The end of the lance was angled 45 degrees inward to provide better penetration of the flame by the secondary air. Adjustment of the insertion depth up to 1.52 m (5 ft) was provided by a locking collar outside the furnace floor.

Nine tests with natural gas fuel and ten with No. 6 fuel oil were conducted to evaluate the effect of staged combustion on NO emissions and burner performance. The first tests with natural gas consisted of varying the injection depth for the staged air. These tests showed no significant reduction in NO with injection heights greater than 1.22 m (4 ft) approximately.

NO emissions increased significantly as the burner air-to-fuel ratio (i.e. the A/F of the fuel-rich zone) was increased when firing natural gas. This trend is shown in Figure 5 as well as the dependence of NO emissions on overall excess oxygen. The tests at low overall excess O_2 reduced NO emissions as much as 67% lower than the conventional burner baseline emissions. At normal O_2, NO emissions were 46% below the baseline value.

The staged combustion modification was also evaluated with the burner firing No. 6 fuel oil. The staged air lances were kept at a fixed insertion depth of 1.22 m (4 ft) for all tests with oil. A similar effect of burner ϕ on NO_x emissions was observed for the tests with No. 6 oil as is shown in Figure 6. At the normal O_2 condition, a reduction of 35% was achieved. With the unit operating in the low O_2 mode a reduction of 51% from the baseline condition was achieved.

An alternative method of producing staged combustion was developed from the cold flow tests described in an earlier section. This technique employed a central cylinder which introduced the secondary air into the flame zone after the primary combustion zone. This modification was tested firing only natural gas.

For this modification the orifice plate was removed from a DBA-16 burner (a conventional burner differing only in tile design from the MA-16 burner) and a 19.1 cm (7-1/2 in.) I.D., 21.6 cm (8-1/2 in.) O.D. cylinder placed in the burner such that its longitudinal axis coincided with the vertical centerline of the burner. The bottom of the cylinder rested on the base of the secondary air section of the burner. Thus, all of the primary air flow (approximately 1/3 of the total air flow) was routed through the cylinder and the rest through the secondary air registers.

Several different cylinder lengths were tried in this series of tests. The height of the top of the cylinder above the gas tips was varied from 7.6 cm (3 in.) to 109 cm (43 in.). The data showed that an optimum height above the gas tips lies between 23 cm (9 in.) and 94 cm (37 in.). The lowest NO_x emission (88 ppm) occurred at 94 cm (37 in.) above the gas tips.

At a cylinder height of 109 cm (43 in.), excess oxygen was varied from 4.9% to the CO limit of 0.5% (with a CO concentration of 439 ppm). At 1.2% excess O_2, the NO_x concentration was 66 ppm, a reduction of 50% from the corresponding O_2 point for the standard DBA-16 burner. At the CO limit, the NO_x level dropped to 54 ppm for a reduction of 59% from the CO limit NO_x concentration emitted by the standard burner.

Low NO_x Burner (Tertiary Air Injection)

Tests were made on a low NO_x burner similar to the conventional MA-16 burner but incorporating a tertiary air register above the primary and secondary air registers. Figure 7 is a schematic of this burner.

Baseline NO_x measurements firing natural gas were about 100 ppm. Excess O_2 was varied from 4.1% down to 2.1%. CO concentration at the minimum O_2 was 47 ppm. NO_x at that O_2 setting was 76 ppm, a reduction of 24% from the tertiary air burner baseline concentration and 30% below the average MA-16 burner baseline emission. The results of these tests are shown in Figure 2.

Firing rate changes were also made firing natural gas. At 100% of capacity (6.5×10^6 Btu/hr) NO_x emission was 155 ppm and dropped to 109 ppm at 37% of capacity. A series of air register adjustments were made at approximately 3% O_2 with the tertiary air burner but produced no appreciable reduction in NO_x levels.

The effect of furnace temperature on NO_x emissions for natural gas fuel was also investigated. The NO_x level tended to rise until a stack temperature of about 1200 K (1700 °F) was attained. Since many tests were conducted with stack temperatures less than 1200 K due to the length of time required for furnace heat-up (about 4 hours) some temperature-related effects were unavoidable in the data. However, the effects were fairly small and were also minimized where possible by conducting a related series of tests (e.g. different excess O_2 points) over the shortest time possible and making baseline checks periodically during the day. There was no large temperature effect on NO_x emissions firing No. 6 oil.

Tests on No. 6 oil with the tertiary air burner consisted of excess O_2 variation and air register adjustments. An oil tip with a 40 deg. spray angle was used for all of the tests with No. 6 oil. The effect of excess O_2 on NO_x emissions for the tertiary air burner using No. 6 oil is shown in Figure 3. The curve is fairly flat showing baseline NO_x emissions to be 272 ppm dropping to 235 ppm at 0.5% O_2, for a reduction of 14%. The CO level at 0.5% O_2 was 57 ppm. These baseline NO_x values were approximately 15% less than the baseline NO_x emissions from the MA-16 burner with a 40 deg. spray angle tip.

The air register setting which produced the lowest NO_x had the primary air register 10% open and the secondary and tertiary air registers 100% open. At 2.9% O_2, NO_x emission at this register setting was 200 ppm, a reduction of 26% from tertiary air burner baseline or 42% from the average MA-16 burner baseline. The variation of excess O_2 at this register setting could not be completed because of a severe coking problem due to the oil tip being placed 1/4-inch too low in the burner throat.

A few tests on the tertiary air burner were also conducted using a shale oil of high nitrogen content (2.1% by weight). Excess O_2 changes coupled with relatively minor register adjustments were made. NO emissions varied from 526 ppm at 6.5% O_2 to 200 ppm at 0.35% O_2. The CO concentration at the latter O_2 was greater than 2000 ppm. At an excess O_2 of 1.2%, NO emission was 295 ppm, or 33% less than the emission at 3.2% O_2 (439 ppm). Also, at the latter point all registers were 50% open, increasing the draft and lengthening the flame. With the primary air register nearly closed, the secondary air register 25% open, and the tertiary air register 100% open, the NO at 3% O_2 was 329 ppm; 25% less than the 439 ppm measured at 3.2% O_2 with all registers 50% open.

Low NO_x Burner (Recirculating Tile)

A low NO_x burner incorporating a self-recirculating tile was evaluated in the research furnace. A special tile was used to achieve some recirculation of fuel vapors and the products of combustion in the immediate vicinity of the burner. The recirculation of these gases is intended to lower the flame zone temperature and, thus, lower thermal NO_x. The NO_x level firing natural gas had a baseline value of 104 ppm (corrected to 3% O_2, dry), only slightly lower than the baseline NO_x found on the conventional MA-16 burner. Excess O_2 was varied from 4.4% down to 0.6%, at which point CO appeared; NO_x emission decreased by 20% to 83 ppm. The CO concentration at 0.6% O_2 was 44 ppm.

Baseline NO_x emissions (at 4% excess O_2) firing No. 2 oil were 110 ppm. Excess O_2 was varied from 5.1% down to 0.5% (CO was 147 ppm at 0.5% O_2). The lowest NO_x emission occurred at an excess O_2 of 1.4% and was 98 ppm, down 11% from the baseline value. Further reduction of the excess O_2 appeared to have no significant result on NO_x emissions.

Flue Gas Recirculation (FGR)

Flue gas recirculation has been demonstrated to be an effective method of NO_x reduction for industrial boilers. Until these tests, flue gas recirculation had not been applied to process heaters for NO_x reduction. It was not possible to duct actual flue gases from the stack to the burner because a high temperature fan was not available. In order to simulate FGR, an auxiliary burner was installed which exhausted into a combustion air duct leading to the burner plenum. A gas-gas heat exchanger was installed to control the temperature of the combustion air-flue gas mixture. The percentage of recirculated flue gas was varied by adjusting the firing rate for the auxiliary burner. Flue gas recirculation rates were varied up to a maximum of approximately 40% when firing natural gas and No. 6 oil.

The effect of FGR rate firing natural gas is shown in Figure 8. The flue gas recirculation is defined by the following expression:

$$\%\ \text{FGR} = \frac{\text{Recirc. mass flow rate x 100}}{\text{Combustion air flow + Recirc. mass flow + Fuel flow}}$$

A reduction in NO_x of 59% from the baseline condition was achieved with approximately 40% FGR at the normal O_2 level ($\sim$ 3%). The overall O_2 level was reduced until the CO limit was reached. This limiting value of excess O_2 was 0.7% O_2. A reduction in NO_x of 63% was measured with the combination of 40% FGR and low O_2 operation.

Figure 9 presents NO as a function of FGR rate for No. 6 oil firing. FGR alone resulted in a reduction of 31% at the maximum recirculation rate ($\sim$ 40%). The combination of FGR and low O_2 operation yielded a reduction of 39% in NO_x emissions.

Steam Injection

The effect of steam injection on NO emissions was evaluated for natural gas firing using two techniques. In the first method, steam was injected into the gas manifold of an MA-16 burner and the steam/gas mixture was then injected radially inward through the standard gas tips. The steam flow rate was varied up to a maximum of 0.0098 kg/sec (78 lb/hr). The effect of steam injection flow rate on NO emissions is shown in Figure 10. The maximum reduction in NO occurred with the maximum steam flow rate. NO emissions were reduced 33% from the baseline condition at 0.0098 kg/sec (78 lb/hr) flow rate.

Since steam for fuel oil atomization is already supplied to the oil gun, injection through the oil gun is a simpler modification than steam injection through the gas tips. This second method of steam injection was tried with a DBA-16 burner. It was hoped that by experimenting with the positioning of the oil tip relative to the gas tips, NO_x emissions could be reduced below the levels of the previous tests.

Maximum reduction in NO_x was achieved at the highest rate of steam injection--114 ppm at 0.0095 kg/s (75 lb/hr) steam flow. Very little difference in NO_x production was observed at the other steam flow rate used (0.0067 kg/s or 53 lb/hr). Thus, the lowest NO_x emissions for steam injection through the oil gun was 13% less than the normal, baseline (3% O_2) NO_x levels for the DBA-16 burner. Thus, the influence of steam injection on NO_x emissions was not nearly as strong for steam injection through the oil gun as it was for steam injection through the gas tips.

Altered Fuel Injection Geometry (AIG)

Previous work with boilers has shown that NO_x emissions can be reduced by altering fuel injection geometry to produce locally fuel-rich zones in the flame. The off-stoichiometric combustion results in lower flame zone temperatures and, thus, lower overall NO_x production. Based on the results of cold flow tests in KVB's laboratory the fuel injection geometry was modified for a DBA-16 natural draft burner.

In the first test series standard gas tips were installed in the DBA-16 with the center firing port facing radially outward rather than inward as is standard practice. On the basis of the cold flow test results discussed earlier, this tip orientation was expected to delay mixing of fuel and air, thus producing a longer, less intense flame and lower NO_x emissions.

The tests showed that NO_x emissions were indeed lower for this tip configuration than for the standard configuration. At 3% excess O_2, the NO_x concentration with outward-facing firing ports was approximately 94 ppm (dry, corrected to 3% O_2), about 31% lower than the NO_x emissions from the standard tip orientation. At an excess O_2 of 1.1%, the NO_x level was 78 ppm. The CO limit occurred at 0.5% O_2, compared with 0.3% O_2 for the standard configuration with a CO concentration of 615 ppm. NO_x at this point was down to 73 ppm, 29% below the standard-configuration value at the CO limit.

The flame shape with the reverse tip orientation was shorter than the normal flame and segmented into four fuel rich regions, one above each of the gas tips. The flame was quite lazy at low firing rates.

Summary of Hot-Firing Test Results

Table II shows that the largest percent reductions in NO_x occurred with staged air or flue gas recirculation techniques. With SCA, these reductions seem to be a relatively strong function of excess air whereas with FGR they are a rather weak function of excess air.

The percent reductions in NO_x emissions observed for modifications to the DBA-16 burner are expected to occur for the same modifications to the MA-16 burner and vice versa with the possible exception of AIG* (where the difference in burner tiles may play an important role in the mixing patterns resulting from the modified injection scheme).

The simplest modifications studied other than LEA were the central staging cylinder and altered fuel injection geometry. AIG as implemented in this test program applies only to gas-firing situations and produces a flame which may be undesirable for practical application. The central cylinder technique produced large percent reductions in NO_x, increased furnace efficiency, and is one of the simplest modifications to implement. Although tests with the staging cylinder were done only on natural gas fuel, it may also be possible to use the cylinder in fuel oil applications provided a coke build-up on the cylinder walls can be avoided.

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*altered fuel injection geometry

SECTION 4

COST EFFECTIVENESS OF COMBUSTION MODIFICATIONS TO NATURAL DRAFT PROCESS HEATER

Summary

The cost effectiveness of the combustion modifications applicable to natural draft process heaters has been evaluated and the results are summarized in Table V. All costs are based on 1978 dollars.

The largest and smallest heater sizes chosen for the present study [147 MW (500×10^6 Btu/h) and 2.9 MW (10×10^6 Btu/h), respectively] represent the two extremes in firing rate for refinery process heaters. The intermediate size of 73.3 MW (250×10^6 Btu/h) was chosen for this cost analysis because it is the current size limit above which steam boilers are regulated by federal emission standards (which do not presently include process heaters).

The total annualized cost per 10^3 kg of NO_x reduction shown in Table V was determined by amortizing the initial fixed capital costs at 20% (corresponding to straight-line depreciation of the capital equipment over 12 years, and assuming a 10% cost of capital, state and federal taxes totalling approximately 11%, and insurance charges of 0.5%). The annual capital charge was added to the annual operating and maintenance cost to obtain the total annualized cost. These costs are shown in Table VI. (Annual operating costs did not include projected fuel savings or costs resulting from modifications for reasons explained below.) The total annualized cost was then divided by the annual reduction in NO_x emissions to obtain the cost effectiveness values in Table V.

The annual reduction in NO_x emissions was calculated for each modification from the maximum percent NO_x reduction listed in Table II using the formula:

$$\text{Annual } NO_x \text{ reduction } (10^3 \text{ kg}) = \frac{\%\ NO_x \text{ reduction}}{100} \times \begin{array}{l}\text{Average baseline emissions} \\ \text{from conventional burner} \\ \text{(ng/J)}\end{array}$$

$$\times \text{ heat input rate (W)} \times 31.536 \times 10^6 \text{ Sec/y} \times 0.8 \text{ (use factor)} \times \frac{10^{15} \text{ ng}}{10^3 \text{ kg}}$$

NO_x emission reductions were determined relative to the conventional MA-16 burner baseline for the following modifications:

1. Lowered Excess Air (LEA)
2. Flue Gas Recirculation (FGR)
3. Staged Combustion Air - Floor Lances (SCA-L)
4. Steam Injection (STM)
5. Tertiary Air Burner (TAB)

NO_x emission reductions were determined relative to the conventional DBA-16 burner baseline for these modifications:

1. Altered Fuel Injection Geometry (AIG)
2. Staged Combustion Air - Central Cylinder (SCA-C)

Although efficiency changes associated with each modification were calculated, these values are not appropriate for estimating annual fuel costs or savings. They are useful only inasmuch as they indicate expected trends in fuel consumption. This is so because the research heater tested by KVB at Location 1 had no process tubes and, therefore, the data do not reflect any inefficiencies or variabilities due to changes in heat transfer to a process stream.

Table V shows that the simplest modifications are the most cost effective. The least expensive modifications, AIG and SCA-C, were tested only in gas-firing application. It is possible that both techniques may be adapted to handle oil-firing applications as well. The more involved modifications, FGR and SCA-L, are less cost effective although they produced the largest percent NO_x reductions.

Most modifications result in lower costs per metric ton of NO_x removed as heater size increases. Only STM and TAB cost effectiveness ratios appear to be relatively independent of size. For the other modifications, both on natural gas and No. 6 oil-firing, the cost effectiveness decreased as heater size increased from 2.9 MW (10×10^6 Btu/h) to 73.3 MW (250×10^6 Btu/h) according to the relation

$$\frac{\text{CE at 73.3 MW}}{\text{CE at 2.9 MW}} = \left(\frac{73.3}{2.9}\right)^a$$

where $-0.67 \leq a \leq -0.47$, $\bar{a} = -0.56$, and S (standard deviation) = 0.07.

Since

$$CE \propto (\text{rated capacity})^{-0.56}$$

and since we expect that

$$NO_x \text{ reduction (metric tons)} \propto (\text{rated capacity})^{1.0}$$

therefore,

$$\text{total annualized cost} \propto (NO_x \text{ reduction}) \times (CE) = (\text{rated capacity})^{0.44}$$

For example, using the total annualized cost for a FGR system for a 2.9 MW (10×10^6 Btu/h) heater given in Table VI at $4300, one can calculate approximately the total annualized cost of FGR for a 73.3 MW heater as follows:

$$\left(\frac{73.3}{2.9}\right)^{0.44} \times (4300) = \$17{,}810$$

Conclusions

The most cost effective combustion modification for NO_x reduction in natural draft process heaters appears to be staged combustion air. The central cylinder technique is the least expensive type of staged air modification, although the largest percent NO_x reduction was obtained using the floor lance technique. Optimization of the central cylinder concept may further improve its NO_x reduction potential, however.

FGR is an effective but more costly modification. TAB, AIG, and STM are all moderately effective in reducing NO_x. STM costs were the highest of any modification for large heater sizes. AIG in the present form applies only to gas fired units, although the concept is adaptable to oil firing. TAB is currently available, represents moderate NO_x reduction capability at moderate cost, and appears more cost effective for smaller heaters firing No. 6 oil.

SECTION 5

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11. Private communication from Vern Sharpe, Sharpe Heating and Ventilating, Alhambra, CA to R. J. Tidona (KVB), June 22, 1978.

12. Private communication from Industrial Gas Engineering, Westmont, IL, to R. J. Tidona (KVB), June 22, 1978.

13. Typical Electric Bills, 1977, Federal Power Commission, FPC R90.

14. Private communication from refinery heater burner manufacturer to S. S. Cherry (KVB), March 21, 1978.

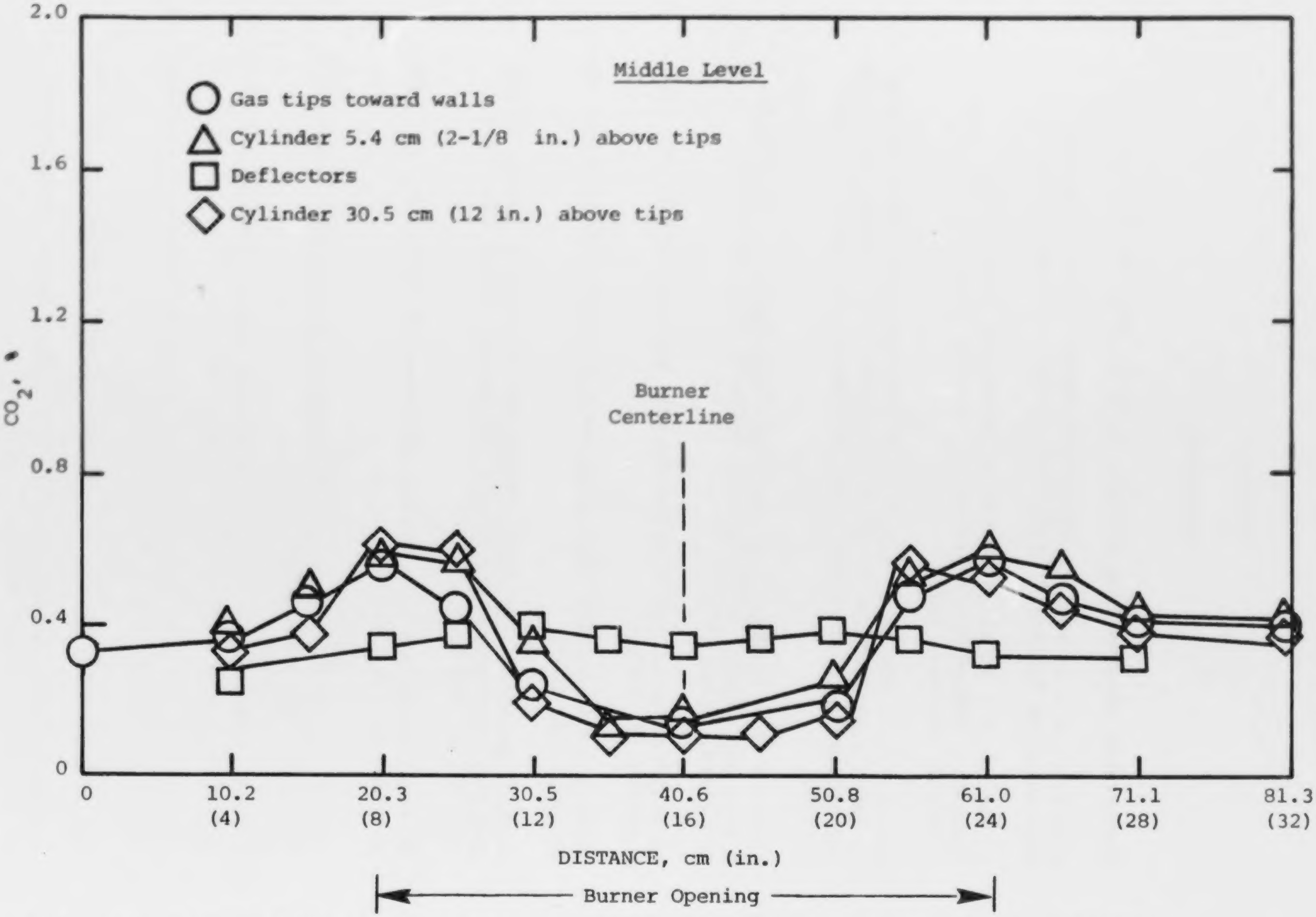

Figure 1. CO_2 concentration vs. centerline distance at one axial position with four different modifications.

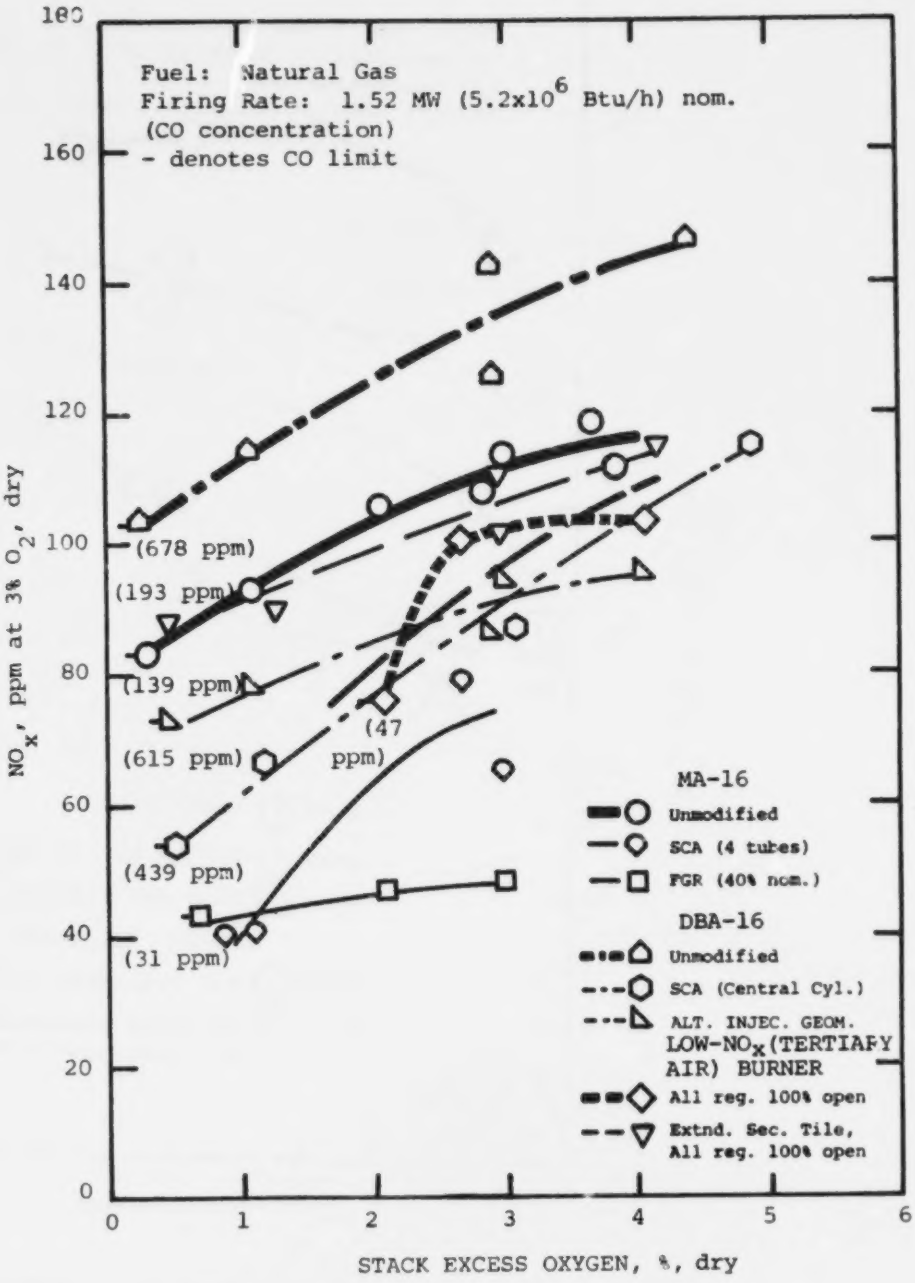

Figure 2. Summary of NO_x emissions as a function of excess oxygen for subscale natural draft furnace firing natural gas.

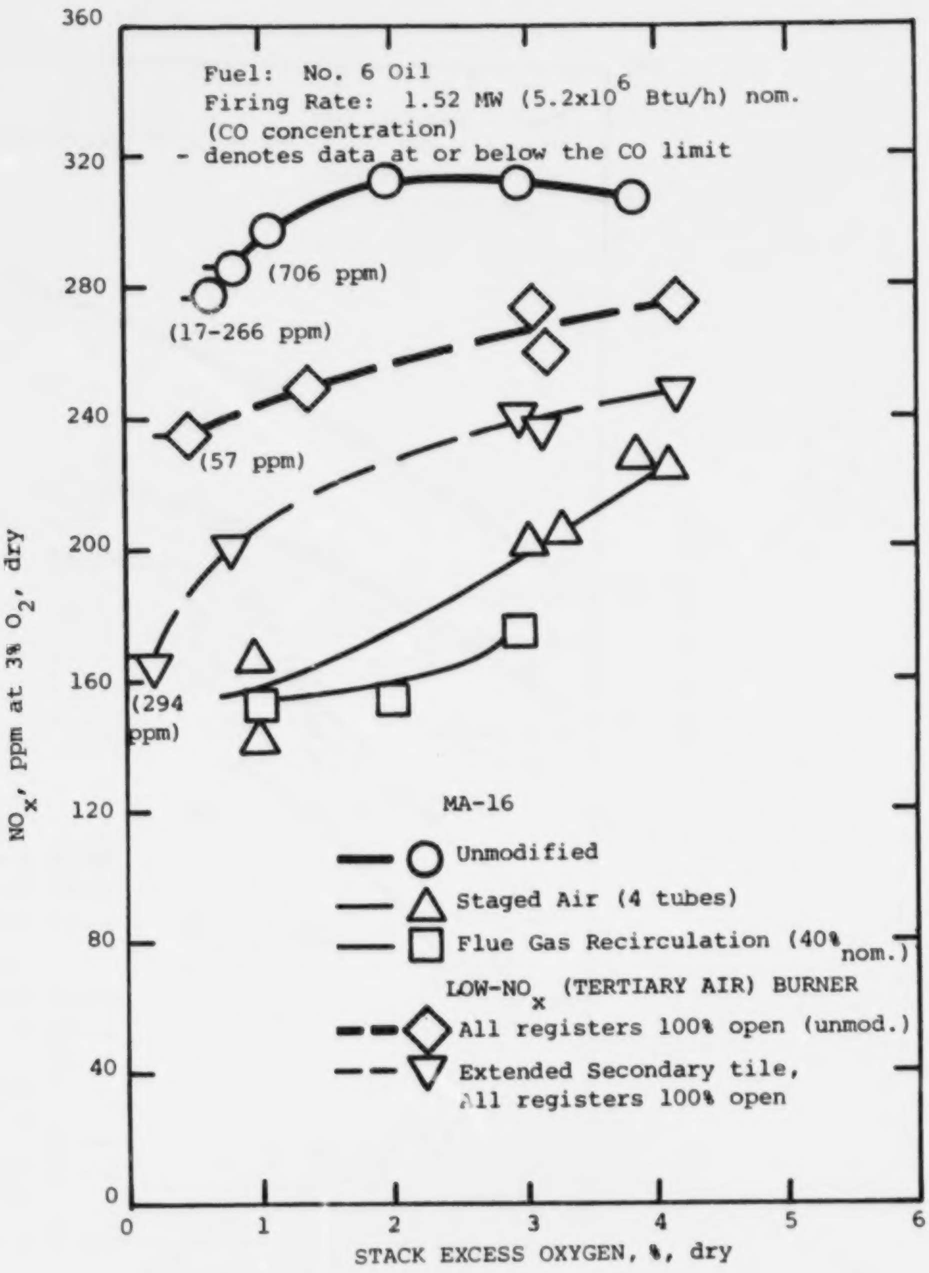

Figure 3. Summary of NO$_x$ emissions as a function of excess oxygen for subscale natural draft furnace firing No. 6 oil.

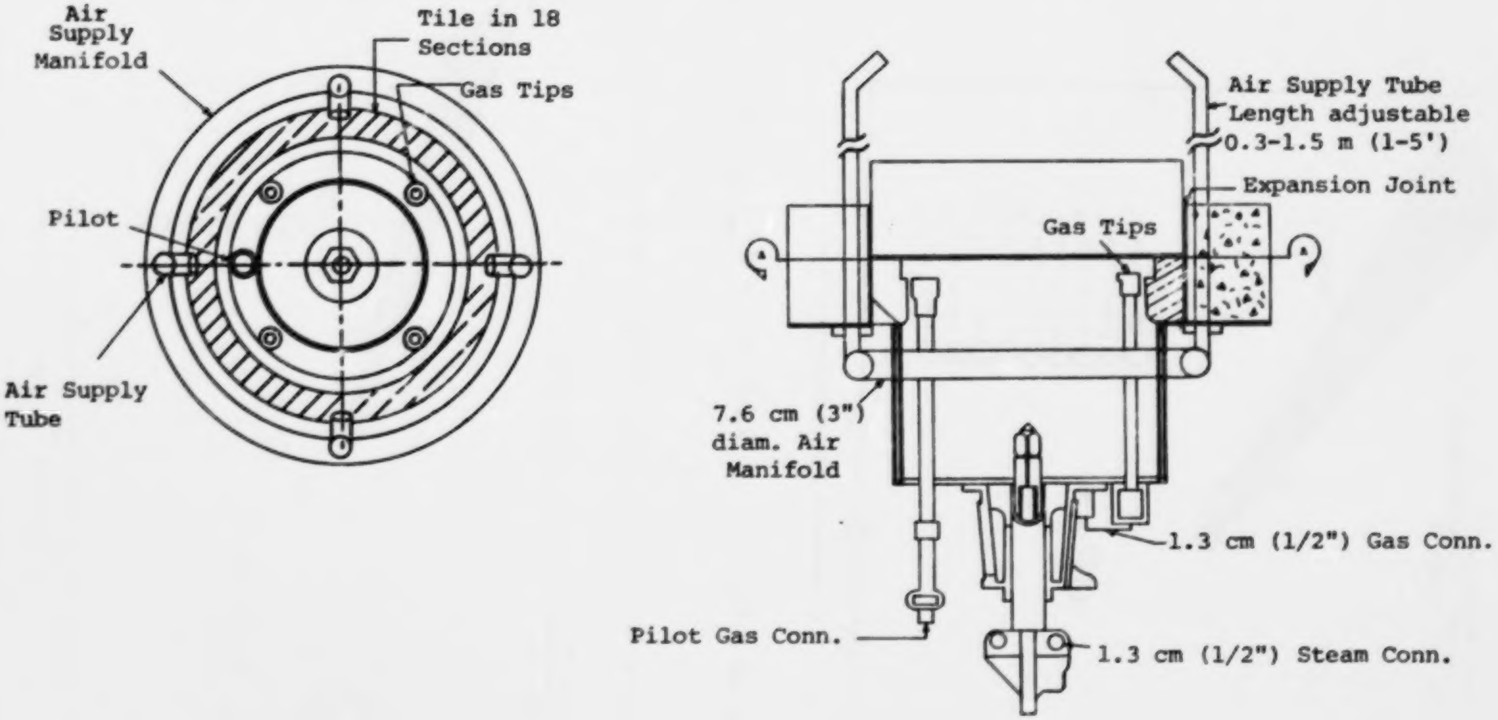
Air Supply Manifold
Tile in 18 Sections
Gas Tips
Pilot
Air Supply Tube
Air Supply Tube Length adjustable 0.3-1.5 m (1-5')
Expansion Joint
Gas Tips
7.6 cm (3") diam. Air Manifold
1.3 cm (1/2") Gas Conn.
Pilot Gas Conn.
1.3 cm (1/2") Steam Conn.

Figure 4. Schematic of staged combustion using an MA-16 burner.

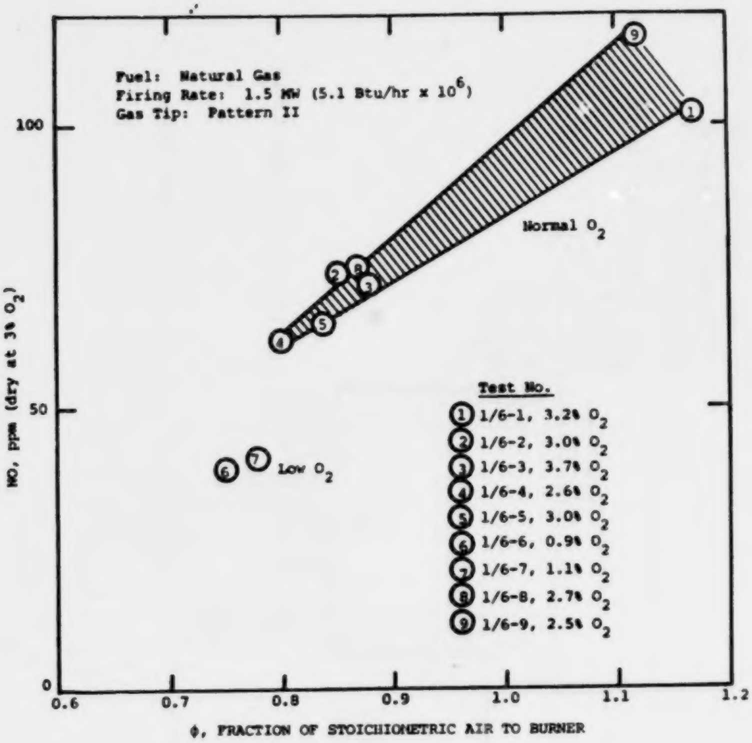

Figure 5. NO emissions as a function of burner ϕ, $(A/F)_{actual}/(A/F)_{stoich}$.

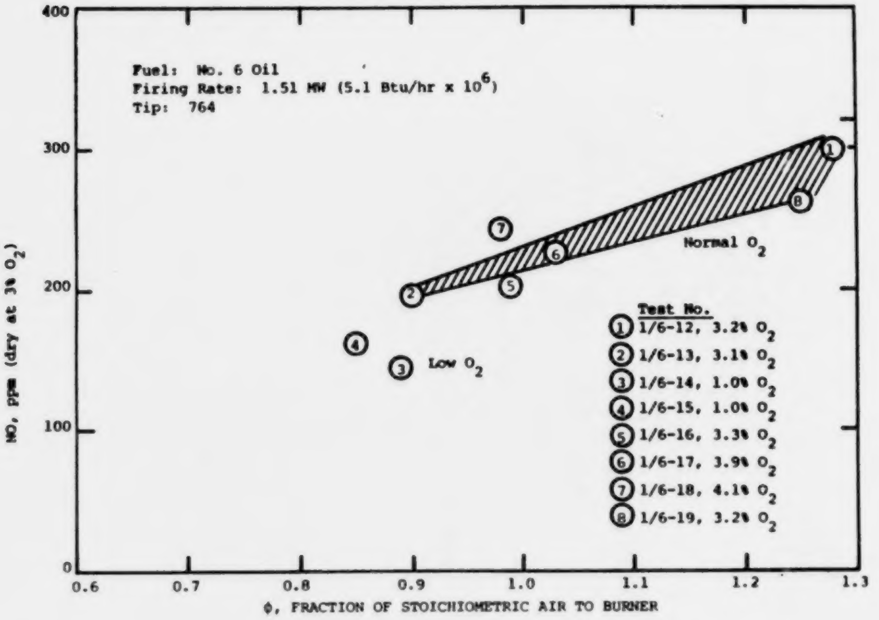

Figure 6. NO emissions as a function of burner ϕ for No. 6 oil firing.

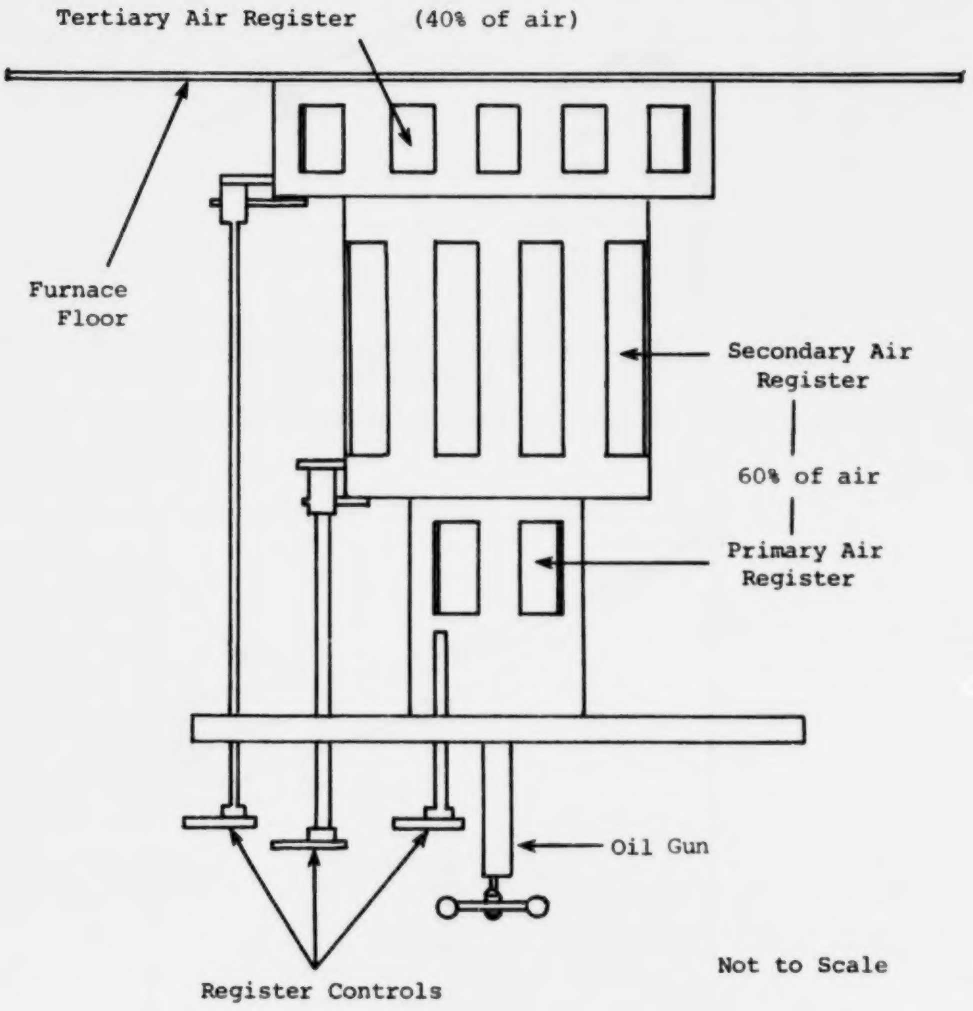
Tertiary Air Register (40% of air)
Furnace Floor
Secondary Air Register
60% of air
Primary Air Register
Oil Gun
Register Controls
Not to Scale

Figure 7. Schematic of tertiary air burner for natural draft process heater.

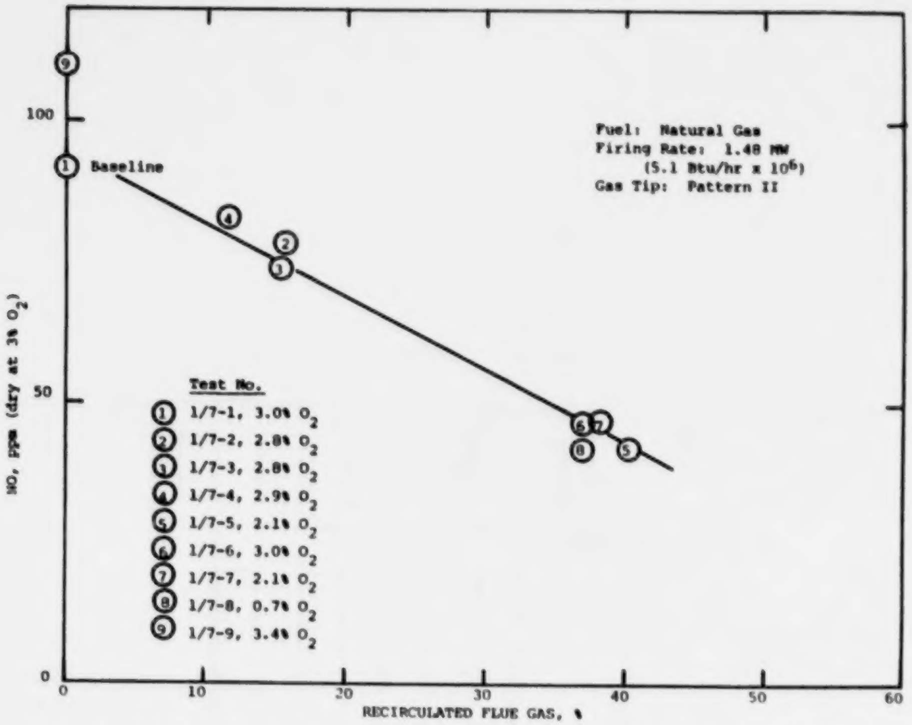

Figure 8. The effect of flue gas recirculation on NO emissions (natural gas).

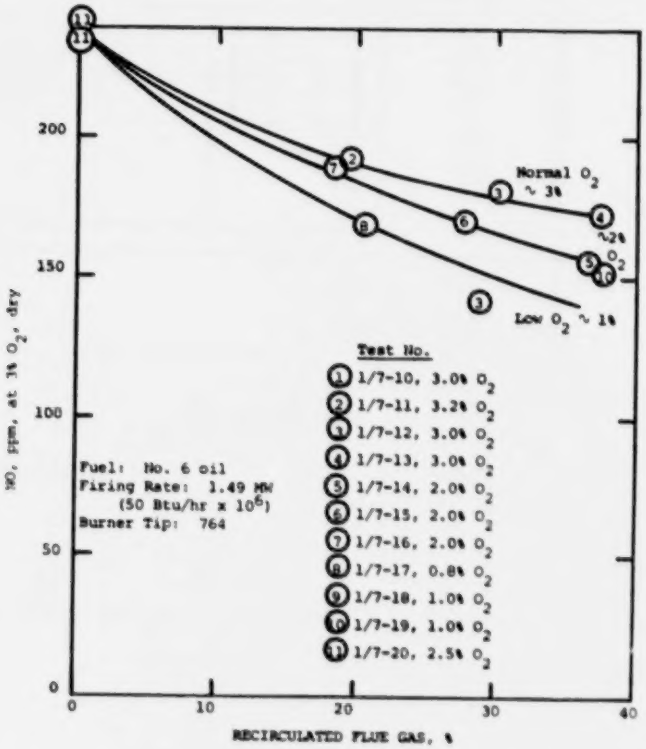

Figure 9. The effect of flue gas recirculation on NO emissions (No. 6 oil).

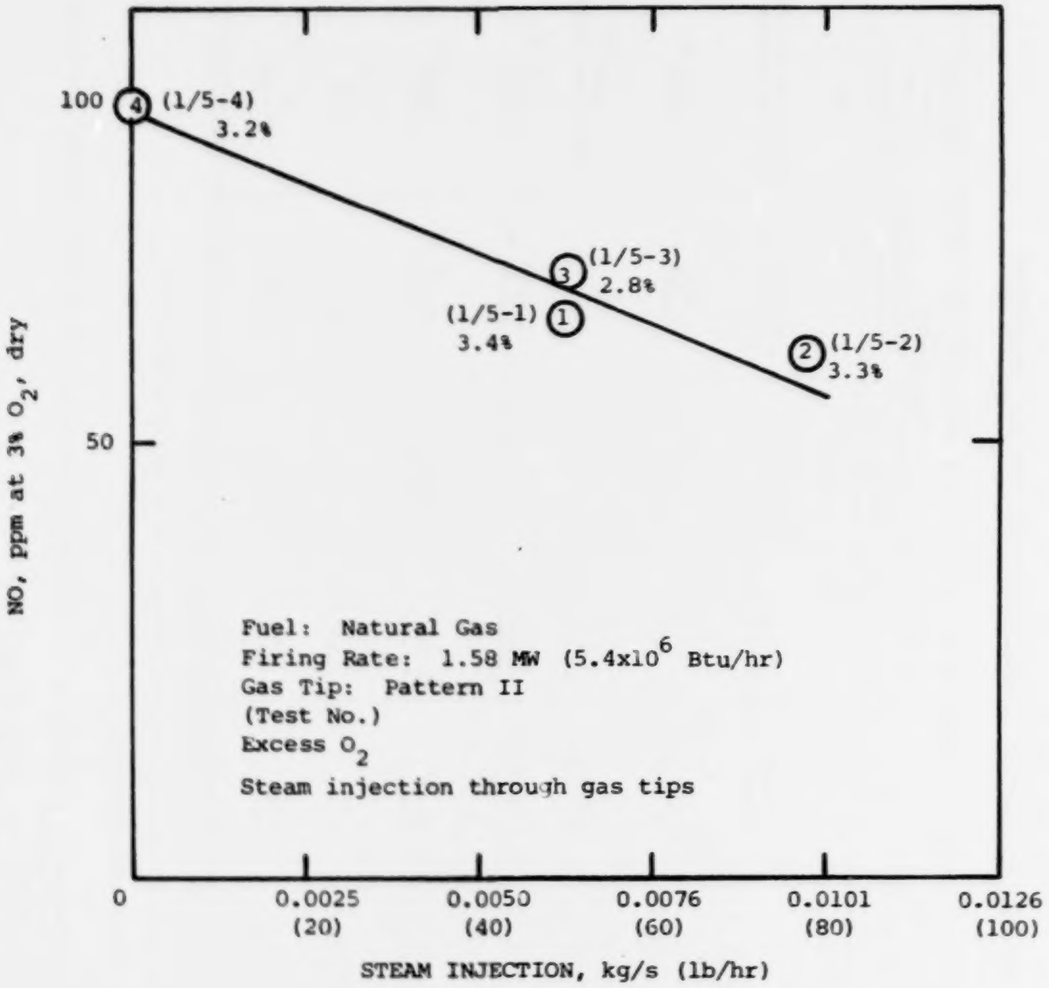

Figure 10. The effect of steam injection on NO emissions for the MA-16 burner firing natural gas.

TABLE I. INDUSTRIAL PROCESS CHARACTERISTICS AND NO_x EMISSIONS

Source Category	Refinery Process Heaters		Glass Container Furnaces	Cement Kilns	Steel Furnaces		Wood/Bark Boilers
Subheading (Reference)	Natural Draft (1)	Forced Draft (1)	All (2)	All (KVB Analysis)	Soaking Pits (KVB Analysis)	Reheat Furnaces (KVB Analysis)	All (j) (KVB Analysis)
Number of units in U.S.	~5400	~600	334	412 (in 1975)	1435 (d)	1264 (d)	~ 1700
Average design capacity per unit, SI (customary)	---	---	1.57 kg/s (150 t/d)	6.60 kg/s (629 t/d)	5.0 kg/s (20 t/h)	25.2 kg/s (100 t/h) (e)	18.8 MW heat absorbed (64,000 lb steam/h)
Average design heat input rate per unit MW(Btu/h)	8.06 (27.5×10^6)	11.1 (38×10^6)	13.1 (44.6×10^6)	39.8 (136×10^6)	7.3 (f) (25×10^6)	88.0 (300×10^6)	23.5 MW (80×10^6)
Total actual annual production kg (tons)	---	---	12.656×10^9 (13.953×10^6)	84.8×10^9 (93.5×10^6)	----	---	328×10^9 (k) (364×10^6 tons steam)
Average heat input per unit of throughput (J/kg (Btu/t)	---	---	8.29×10^6 (a) (7.14×10^6)	6.04×10^6 (b) (5.2×10^6)	1.45×10^6 (1.25×10^6) (g)	3.48×10^6 (3.0×10^6) (g)	2.8×10^6 (l) (2.4×10^6)
Total annual heat input J (Btu)	1.37×10^{18} (1.3×10^{15})	211×10^{15} (200×10^{12})	105×10^{15} (99.6×10^{12})	513×10^{15} (486×10^{12})	185×10^{15} (h) (175×10^{12})	353×10^{15} (h) (335×10^{12})	915×10^{15} (873×10^{12})
Average emission factor ng/J (lb/10^6 Btu)	68.8 (0.16)	133.3 (0.31)	370 (0.861)	1372 (3.19) (c)	56 (0.13) (i)	52 (0.12) (i)	156 (m) (0.36)
Total NO_x emission Gg/y (t/y)	93.4 (103,000)	28.1 (31,000)	38.89 (42,900)	704 (776,000)	10.32 (11,375)	18.23 (20,100)	141 (157,000)

NOTES:
(a) based on unpublished data from California Air Resources Board study of SOx emission by KVB.
(b) from Reference 3
(c) average of emission factors determined from two field tests conducted by KVB as reported in Reference 4.
(d) from Reference 5
(e) from Reference 6 (average for slab reheat furnaces)
(f) from Reference 7
(g) from References 3, 6 and 7
(h) from Reference 3 (total annual heat input for both soaking pits and reheat furnaces combined) and Reference 8 (fraction of total heat input to soaking pits, fraction to reheat furnaces)
(i) from KVB tests as reported in Reference 4
(j) based on 1977 NEDS point source listing (Ref. 9) with KVB emission factor
(k) assumes operation at 80% rated load for 11 months out of the year
(l) assumes heat requirement of 1200 Btu/lb of steam
(m) from Reference 4; average for wood + coal and wood + NG boilers.

TABLE II. RESULTS OF TESTS ON A SUBSCALE NATURAL DRAFT PROCESS HEATER

		Average	Baseline NO_x	
Modification	Fuel	ng/J	ppm, dry at 3% O_2	Maximum Percent Reduction in NO_x
Lowered Excess Air	Natural Gas	58	113	27
Lowered Excess Air	No. 6 Oil	174	311	10
Staged Combustion (Floor Lances) at normal excess oxygen	Natural Gas	61.2	120	46
Staged Combustion (Floor Lances) at low excess oxygen	Natural Gas	61.2	120	67
Staged Combustion (Floor Lances) at normal excess oxygen	No. 6 Oil	172	307	35
Staged Combustion (Floor Lances) at low excess oxygen	No. 6 Oil	172	307	51
Staged Combustion (Central Cylinder) at normal excess oxygen	Natural Gas	66.8	131	31
Staged Combustion (Central Cylinder) at low excess oxygen	Natural Gas	66.8	131	59
Tertiary Air Burner, Lowest NO_x Configuration	Natural Gas	54.6	107	30
Tertiary Air Burner, Lowest NO_x Configuration	No. 6 Oil	160	285	42
Recirculating Tile Burner, Lowest NO_x Configuration	Natural Gas	54.6	107	3
Recirculating Tile Burner, Lowest NO_x Configuration	No. 2 Oil	61.8	112	13
Flue Gas Recirculation at normal excess oxygen	Natural Gas	59.2	116	59
Flue Gas Recirculation at low excess oxygen	Natural Gas	59.2	116	63
Flue Gas Recirculation at normal excess oxygen	No. 6 Oil	141	252	31
Flue Gas Recirculation at low excess oxygen	No. 6 Oil	135	241	39
Steam Injection	Natural Gas	54.6	107	33
Altered Fuel Injection Geometry at normal excess oxygen	Natural Gas	66.8	131	31
Altered Fuel Injection Geometry at low excess oxygen	Natural Gas	66.8	131	44

TABLE IV. SUMMARY OF AVERAGE BASELINE GASEOUS EMISSIONS FOR UNMODIFIED BURNERS

	Heat Input Rate MW	(10^6 Btu/h)	O_2 %	CO_2 %	NO_x ppm*	NO_x ng/J	NO ppm*	NO ng/J	CO ppm*	SO_2 ppm*
Natural Gas										
MA-16	1.53	(5.2)	3.0	10.7	107	54.6	103	53.8	0	0
DBA-16	1.52	(5.2)	3.0	10.3	131	67.0	127	64.9	0	0
Low-NO_x Burner (Tertiary Air Injection)	1.49	(5.1)	3.2	10.4	92	47.1	87	44.4	0	0
Low-NO_x Burner (Recirculating Tile)	1.47	(5.0)	3.1	9.9	104	53.0	104	53.0	0	0
No. 6 Oil (0.3% N)										
MA-16	1.47	(5.0)	3.0	13.3	285	159	278	156	0	1015
Low-NO_x Burner (Tertiary Air Injection)	1.43	(4.9)	3.1	13.7	265	149	261	147	0	1334
No. 2 Oil (0.01% N)										
MA-16	1.41	(4.8)	3.1	12.7	112	63	108	61	0	46
Low-NO_x Burner (Recirculating Tile)	1.49	(5.1)	3.9	12.6	110	61.7	105	58.9	0	38

*Corrected to 3% O_2, dry

TABLE V. COST EFFECTIVENESS (\$/10^3 kg of NO_x reduction) OF COMBUSTION MODIFICATIONS TO A NATURAL DRAFT PROCESS HEATER (NOT INCLUDING ANNUAL FUEL COSTS/SAVINGS)

Modification	Heater Size					
	2.9 MW ($10x10^6$ Btu/h)		73.3 MW ($250x10^6$ Btu/h)		147 MW ($500x10^6$ Btu/h)	
	Natural Gas	No. 6 Oil	Natural Gas	No. 6 Oil	Natural Gas	No. 6 Oil
Low Excess Air	0	0	0	0	0	0
Altered Injection Geom. (Normal O_2)	\$6.60	--	\$0.78	--	\$0.78	--
Altered Injection Geom. (with LEA)	\$4.70	--	\$0.55	--	\$0.55	--
Staged Air - Central Cyl. (Normal O_2)	\$200	--	\$43	--	\$33	--
Staged Air - Central Cyl. (with LEA)	$\boxed{\$100}$	--	$\boxed{\$23}$	--	$\boxed{\$18}$	--
Staged Air - Floor Lances (Normal O_2)	\$1000	\$460	\$160	\$70	\$130	\$57
Staged Air - Floor Lances (with LEA)	\$710	\$320	\$110	$\boxed{\$48}$	\$87	$\boxed{\$39}$
Flue Gas Recirculation (Normal O_2)	\$1800	\$1200	\$320	\$200	\$320	\$200
Flue Gas Recirculation (with LEA)	\$1700	\$940	\$300	\$160	\$300	\$160
Steam Injection (no initial cost)	\$990	--	\$970	--	\$960	--
Steam Injection (incl. initial cost)	\$1400	--	\$1100	--	\$1000	--
Tertiary Air Burner	\$250	$\boxed{\$60}$	\$250	\$60	\$250	\$60

☐ Indicates lowest cost effectiveness for each size and fuel, excluding low excess air and altered injection geometry.

TABLE VI. TOTAL ANNUALIZED COSTS (IN $) NOT INCLUDING FUEL COSTS (SAVINGS) OF COMBUSTION MODIFICATIONS TO A NATURAL DRAFT PROCESS HEATER (AMORTIZING INITIAL CAPITAL COSTS AT 20%)

Modification	Heater Size 2.9 MW (10×10^6 Btu/h)	 73.3 MW (250×10^6 Btu/h)	 147 MW (500×10^6 Btu/h)
Low Excess Air	0	0	0
Altered Injection Geom.	10	30	60
Staged Air - Central Cyl.	300	1660	2560
Staged Air - Floor Lances	1900	7330	11800
Flue Gas Recirculation	4300	18800	38100
Steam Injection (if initial installation necessary)	1900	35400	69900
Steam Injection (no initial installation required)	1300	32200	64300
Tertiary Air Burner	300	7500	15000

POLLUTANT EMISSIONS FROM "DIRTY" LOW- AND MEDIUM-Btu GASES

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ABSTRACT

Data were collected to determine the emissions from "dirty" low- and medium-Btu gases when combusted on industrial process burners. The fuels utilized were blended to have the composition typically found for Wellman-Galusha oxygen (WGO) and air (WGA) fuel gases. Base-line data were collected for natural gas, ambient WGO and WGA, and hot WGO (700K) and WGA (616K). Then ammonia, hydrogen sulfide, coal tar and char were added singly at various levels and in combinations to the hot fuels in a parametric study to determine the effects of these contaminants on pollutant emissions. The burners used in this study were a forward-flow baffle burner and a gas momentum controlled kiln burner. These burners were mounted in turn on a pilot-scale test furnace equipped with water tubes as a load and were each fired at 1.03 ± 0.07 MW_t ($3.50 \pm 0.25 \times 10^6$ Btu/hr) with 10% excess air and 477 K (400°F) air preheat.

Based on a detailed analysis of the experimental data, the following conclusions were made:

- Low-Btu fuels not subjected to post-gasifier cleanup can yield NO_x levels greatly above the thermal levels for the clean fuels and for natural gas.

- In turbulent diffusion flames, fuel-NO_x increases with an increase in a) the amount of fuel-nitrogen, b) the amount of fuel-sulfur, c) the level of excess air, and d) the degree of initial fuel/air mixing.

- Attempts to close the fuel-sulfur balance were unsuccessful. Whether this shortfall is due to sampling/instrument effects or large concentrations of some unmeasured sulfur-containing species is not clear. Further work should be done in this area.

- Compared to natural gas, heat transfer to the load is reduced for the low-Btu fuels tested. This heat transfer is not greatly affected by the presence of contaminants (tar and char) at levels characteristic of raw gasifier effluents.

INTRODUCTION

The objective of the research program was to provide and evaluate quantitative data on the differences in the environmental quality of effluent combustion products and furnace efficiency when retrofitting a natural gas/oil industrial burner with "dirty" intermediate- and low-Btu gases. These data were collected from the IGT pilot-scale industrial test furnace.

This program was intended to complement a recently completed EPA program (1), which evaluated the emissions resulting from burning "clean" low- and medium-Btu gases in boilers. This previous work provided quantitative information on emissions using gases processed by ambient temperature sulfur cleanup systems and enabled correlating emissions to gas composition and operating conditions.

Work is currently being done by the government and industry to develop high temperature-post gasification sulfur cleanup systems. The high-temperature cleanup processes leave varying amounts of tars, oils, and ammonia in the product gas stream. Since these contaminants can contribute to emissions resulting from combustion of fuel gas, it is important that the magnitude of the potential problem be evaluated and the results used to determine if high-temperature sulfur removal systems are feasible from an environmental viewpoint.

"Dirty" low- and intermediate-Btu gases could have higher flame emissivities than "clean" gases due to tar-oils and char, which could result in increased furnace efficiency. However, the hot "dirty" fuel often has a lower adiabatic flame temperature due to a higher water content in the fuel. This program was designed to quantify such changes in efficiency and provide data on which a decision can be made on retrofitting an industrial process furnace with low- or intermediate-Btu gases.

This program was designed to provide data for two gases and two burners as a means to broadly scope the potential environmental problem.

ENVIRONMENTAL ASSESSMENT OF COMMERCIAL GASIFICATION PROCESSES

Tars and Oils

A wide variety of coal conversion systems can be used to produce a low- or intermediate-Btu gas. However, the operating conditions of the gasifier and cleanup system will have a significant effect upon the types of potential pollutants present in the off-gas. In most of the gasification processes, the initial treatment of the coal determines the major characteristics of the raw gas. Initial dimensions of contact time, gas-to-coal ratio, contacter types (entrained bed, fluidized-bed, fixed-bed, stirred liquid), and contacter mode (cocurrent, countercurrent, back-mixed reactor), and perhaps other variables should be included for a complete profile of the types of pollutants that might be present in the raw gasifier product, particularly tars and oils.

These liquids would be condensed and removed from the gas stream using ambient temperature sulfur cleanup systems, but would not be removed when using the high-temperature processes. If not condensed, they would be burned during the combustion process with the off-gas. The combustion of these tar- and oil-containing gases in an environmentally acceptable manner was part of the overall experimental evaluation.

The basic character of the complex coal organic structure is aromatic. Therefore, the tars that are expelled from coal during devolatilization in lower temperature reactors may be expected to contain naphthalenes, indenes, anthracenes, and similar compounds. Oxygenated compounds such as phenols and cresylic acids may be expected, in addition to nitrogen- and sulfur-containing ring structures. In moderate temperature reactors, these complex aromatics are hydrocracked and possibly hydrodealkylated to simpler BTX (benzene-toluene-xylene) streams.

Of the conversion processes commercially available (e.g. Koppers-Totzek, Winkler, Wellman-Galusha, and Lurgi), only Wellman-Galusha and Lurgi produce significant amounts of tars and oils.

Sulfur

The sulfur that is present in coal is one of the primary reasons that low- and intermediate-Btu gasification processes are being developed. Much of the coal in this country contains significant quantities of sulfur, and present methods of sulfur oxides reduction, such as stack-gas scrubbing, are not viewed as sufficiently reliable, effective, or operable by many potential coal users. The concept of low- and intermediate-Btu gasification permits the removal of the sulfur from the gas before combustion, and overall sulfur oxides emissions may be significantly reduced utilizing this approach. The other major impetus for the development of low- and intermediate-Btu gasification is the higher overall efficiency achieved when the gas is utilized to produce electricity in combined-cycle operations.

The sulfur that occurs naturally in the coal will be largely driven into the raw product gas although hard data do not exist on the form of the sulfur in the off-gas. Thermodynamically, the great majority of the sulfur should exist as hydrogen sulfide.

Purification systems are available for removing a large quantity of the hydrogen sulfide that is present in the raw product gas. Currently, much work is being done to develop and demonstrate the use of high-temperature sulfur cleanup systems. However, even the most optimistic projections show an H_2S concentration of 100 ppm in the off-gas.

Nitrogen

Nitrogen that is present in coal may be considered as fixed nitrogen. Fixed nitrogen may be defined as nitrogen that is chemically bound to other species in contrast to molecular nitrogen (N_2) that is present in the air. The nitrogen in coal tends to gasify simultaneously with the carbon (2). Generally, the nitrogen is expected to react with the hydrogen during gasification to form ammonia. The existence of ammonia in raw gasifier effluents has been confirmed by many investigators.

Hydrogen cyanide (HCN) is also present in the raw gas effluents. Published data report concentrations of less than 10% of the ammonia concentration. Thus, the major contribution in NO_x formation will be ammonia.

TEST FACILITIES

PILOT-SCALE FURNACE

The experimental work was carried out in the pilot-scale furnace which is 14 feet* long and has a cross-sectional area of 21.3 sq ft. There are 33 panels or "sampling doors" along one sidewall that allow insertion of probes at any axial position from the burner wall to the rear wall. The facility can be used for firing burners rated up to 6 million Btu/hr (6 MBtu/hr). Combustion air temperatures up to 1000°F can be generated with a separately fired air preheater.

The furnace is also equipped with 58 water cooling tubes, each of which can be independently inserted through the roof, along the sidewalls. Varying the number of tubes, their location, and the depth of insertion allows control over the magnitude and character of the load that can be placed on the furnace. The amount of heat absorbed by each tube can be determined by measuring the water flow through each tube and the temperature difference between the inlet and outlet.

In addition to the combustion air preheater, a separately fired fuel preheater is available that can heat 12,000 SCFH of low-Btu gas to any desired temperature up to 800°F. Temperatures up to 1200°F are attainable with lower flow rates.

LOW-Btu GAS GENERATING SYSTEM

The low- and medium-Btu gases are generated using a special gas-generating and fuel-preparation facility that can produce varying ratios of hydrogen and carbon monoxide. Natural Gas, carbon dioxide, and steam are passed through reaction retorts contained in a vertical cylindrical furnace. After compression, the product gas is blended with nitrogen, methane, carbon dioxide, and/or steam, as required, to obtain the specified composition of the fuel gas to be tested.

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* It is EPA policy to use metric units. However, in this report, English units are occasionally used for convenience.

Table I gives the composition of the Wellman-Galusha oxygen (WGO) and Wellman-Galusha air (WGA) fuel gases, which were chosen to be simulated as test gases for the program.

DOPANT SYSTEM

Although a number of potential pollutants might be present in the raw gasifier products, the data now available on their occurrence and/or concentration are poor. Consequently, the trials concentrated on the fate of species for which some data exists (tars, oils, ammonia, particulate, and hydrogen sulfide). These contaminants were "doped" into the hot experimental fuel gas. Although coal-tar does not have an identical chemical analysis to that of tars and oils found in raw off-gas, it does contain all the aromatic and oxygenated compounds that are found. Thus, realistic characterization of the combustion process and pollutant emissions can be anticipated. Analyses of the tar and char are presented in Tables II and III.

Figure 1 gives a schematic diagram of the doping system. Raw coal-tar from a coke oven was used as the tar introduced into the hot gas stream. Ammonia and hydrogen sulfide were blended into the fuel gas stream from cylinders. The flow of these dopants was adjusted using rotameters. The tar, which was a liquid at room temperature, was forced from a container under nitrogen pressure through a nozzle in the hot fuel feed line where it was steam atomized. Char was screw-fed into the hot fuel. Doping rates were controlled by varying the screw speed.

INSTRUMENTATION

The instrumentation used during this study is fully described in EPA report 600/7-78-191. A listing is provided here:

- Suction pyrometer with Pt/Pt-13% Rh thermocouple for gas temperature
- Beckman 742 Polarographic Oxygen (O_2)
- Beckman Paramagnetic Oxygen (O_2)
- Beckman NDIR Methane (CH_4)
- Beckman NDIR Carbon Monoxide (CO)
- Beckman NDIR Carbon Dioxide (CO_2)

- Varian 1200 Flame Ionization Chromatograph (Total CH and C_2 to C_9)
- Beckman NDIR Nitric Oxide (NO)
- Beckman UV Nitrogen Dioxide (NO_2)
- Thermo Electron Pulsed Fluorescent Sulfur Dioxide (SO_2)
- Hewlett-Packard Thermoconductivity Chromatography, Hydrogen (H_2), Nitrogen (N_2), Argon (Ar), CO, CO_2, C_1 to C_5, Oxygen (O_2)
- Beckman Chemiluminescent NO-NO_x
- Molectron PR-200 Radiometer for radiation intensity
- Research Appliances Corp. Model 2414 "Staksamplr" for EPA Method 5 particulate measurements
- Anderson Mark III multistage, cascade impactor for particulate size distribution.

RESULTS

Summaries of all the test data for the ported baffle burner and the fuel momentum controlled burner are presented in Tables IV and V respectively. The following sections present the detailed findings by fuel type.

NATURAL GAS: BASE-LINE TESTS

Baffle Burner Tests

A baffle burner, representative of the forward-flow type, was the first burner to be utilized. The burner tested, illustrated in Figure 2, is full scale and is available as an off-the-shelf item from the manufacturer (Bloom Engineering). The burner consists of a centrally located gas nozzle, surrounded by a high-temperature refractory baffle that has ports for the injection of combustion air into the furnace. The flame patterns produced by this burner can be altered by changing the angles of these air ports or their diameters with the insertion of different baffles.

This type of burner is found on many large-scale industrial process heating furnaces such as steel reheating, batch glass melting, aluminum holding, and tunnel kilns. The baffle design selected for testing produces a flame-to-furnace length ratio equal to the flame-to-preheat section length ratio found in a five-zone steel slab reheat furnace.

The rate of heat absorption and efficiency (approx 35%) of the preheat section of a steel reheat furnace was simulated on the pilot-scale furnace by inserting water cooling tubes along the furnace sidewall.

Fuel Momentum Controlled Burner Tests

A fuel momentum controlled kiln burner (FMCB) was the second burner tested and the pilot-scale test furnace was set up to specifically simulate a cement kiln. The critical operating parameters were a) a length sufficient to simulate the calcining and reaction zones, b) the firing density, and c) the heat absorption profile.

A typical kiln consists of preheat, calcining, reaction, and cooling zones. The cross-sectional area (20 sq ft) and length (14 ft) of the IGT pilot furnace allow for simulation of the two zones of primary importance: the calcining and reaction zones. It is the heat transfer in these zones that is sensitive to fuel type. These two zones occupy about one-third of the overall kiln length, with the calcining zone being twice the length of the sintering zone. The flame usually extends three-quarters of the length of these zones.

Firing densities in rotary kilns range from 10,000 to 20,000 Btu/hr-ft^3. A firing density value of 12,500 Btu/hr-ft^3 was used for these tests and is typical of many kilns. This requires a firing rate of 3.5 million Btu/hr for our test furnace volume. Cement kilns require about 5 to 7 million Btu/ton of clinker. Assuming 6 million Btu/hr, our furnace would simulate a production rate of 1150 lb/hr.

This rate of heat absorption and this efficiency were simulated on the pilot-scale furnace by inserting water cooling tubes along the furnace sidewalls, while firing natural gas at 3.5 million Btu/hr. Figure 3 is a schematic of the kiln burner fuel injector.

WELLMAN-GALUSHA OXYGEN: EMISSION STUDIES

Prior to the doping studies, base-line emissions were obtained for natural gas and Wellman-Galusha oxygen (WGO) fuel gas on the baffle burner and kiln burner and are shown in Table VI. Both fuels were fired at 1.03 ± 0.07 MW_t (3.50 ± 0.25 X 10^6 Btu/hr) with 10% excess air. The above variation in fuel heat input represents the range of firing rates from run to run and not the firing rate fluctuation during a given run, which was minimal. For all tests, the furnace was at positive pressure. All concentrations presented are dry analyses at the flue entrance. For NO_x levels under 100 ppm the reproducibility of the reported values was ± 5 ppm while at the higher levels it was ± 10 ppm. Recent studies (3) have shown that quenching may cause interference in NO_x measurements. Our results, however, have not been compensated for such effects.

With natural gas and WGO on the baffle burner a 0.063-m inside diameter (ID) (nominal 2-1/2 inch Schedule 40) fuel nozzle was used. Fuel velocities at the nozzle exit were 40 m/s (130 ft/s, ambient WGO) and 110 m/s (360 ft/s, hot WGO). Natural gas velocity was 9.5 m/s (28 ft/s). Combustion air entered the furnace at velocities of 19 m/s (63 ft/s) for the low-Btu fuels and 23 m/s (76 ft/s) for natural gas.

On the kiln burner a 0.043-m ID (nominal 1-1/2 inch Schedule 10) axial fuel nozzle with a 0.019-m ID (7/8 inch, 16 gauge) tube as the radial injector was employed for natural gas and ambient (322 K) WGO. For hot (700 K) WGO, a 0.108-m ID (nominal 4 inch Schedule 10) axial fuel nozzle with a 0.064-m (nominal 2-1/2 inch Schedule 10) radial injector was required to achieve comparable fuel velocities as well as to maintain flame stability. The amount of radial flow for natural gas on the kiln burner was 95% of the total. This value was selected to give a stable flame of a size compatible with the furnace dimensions. For both ambient and hot WGO, the flow was 22% radial, chosen to give a flame length comparable to that of natural gas. With the fuels studied, gas injected radially was at sonic velocity while the axial component entered at 85 m/s (280 ft/s) for ambient WGO. The axial velocity was 52 m/s (170 ft/s) for hot WGO on the larger nozzle. With natural gas, the axial flow was 1.2 m/s (4 ft/s). Air velocity with the kiln burner was 3.1 m/s (10 ft/s).

In order to determine the effects of the various potential sources of fuel-NO_x on the measured NO_x levels, char, tar, and ammonia were individually doped into hot WGO. The doping system and the char (0.66 weight percent nitrogen) and tar (0.55 weight percent nitrogen) analyses were presented in the preceeding section. The results of the char and tar doping tests are presented in Tables VII and VIII. For char, the doping rate varied from 0.13 to 0.86 g/s or 0.4 to 2.7 grains/SCF, while the tar levels of 0.36 to 0.73 g/s correspond to 1.1 to 2.3 grains/SCF. The total NO_x levels measured for char or tar are comparable and show a NO_x increase from 0 to 30 ppm. The fact that bound sulfur in the tar (0.47 weight percent sulfur) and char (1.64 weight percent sulfur) is converted to sulfur oxides is evidenced by the SO_2 levels. Increases in NO_x levels above undoped thermal NO_x levels cannot simply be ascribed to fuel-nitrogen conversion when fuel-sulfur is also present. Work of IGT (unpublished) and others (4) has shown that in turbulent diffusion flames thermal NO_x, as well as fuel NO_x, is enhanced by fuel-sulfur. In the tar doping studies the problem of char-like residue collecting inside the fuel nozzle was encountered. On the baffle burner, this residue amounted to about 3% by weight of the doped tar, whereas on the kiln burner the residue was around 7% of the tar input.

The effects of various levels of fuel-nitrogen, in the form of ammonia, on the conversion of fuel-nitrogen to NO_x are shown in Tables IX and X and Figures 4 and 5 for the baffle and kiln burners with 10% excess air. Ammonia was metered into the hot WGO at levels of 0.02 to 1.00 volume percent of fuel input. The oxygen level at the flue was maintained at 1.8 ± 0.1% (dry analysis), corresponding to 10% excess air. The fraction of ammonia converted to NO_x decreases as the fraction of ammonia in the fuel increases. The results are comparable for both burners with the kiln burner giving a somewhat higher conversion.

The effects of changing the amount of excess combustion air are also shown in Tables IX and X and Figures 4 and 5 for the baffle and kiln burners with 20% excess air. Here, ammonia constituted from 0.20 to 1.09 volume percent of the fuel input. The oxygen level at the flue was held at 3.3 ± 0.1% (dry analysis) to keep the excess air level at 20%. A comparison of Figures 5 and 6 shows that the fraction of ammonia converted to NO_x increases with an increase in the availability of oxygen. Again the kiln burner gives a slightly higher ammonia-to-NO_x conversion than the baffle burner.

The effects of ammonia doping on flue oxygen levels were determined by setting the flue oxygen to the level required for 10% excess air with 1.0% ammonia addition and then reducing the ammonia level in five steps to 0.2%. For a given reduction of ammonia the oxygen level rose by an amount consistent with the reaction -

$$NH_3 + (0.75 + \frac{f}{2})O_2 = f\ NO + (\frac{1 - f}{2})N_2 + 1.5\ H_2O$$

where f is the fraction of ammonia converted to NO_x.

Because raw hot gasifier effluents contain both fuel-nitrogen and fuel-sulfur, the effects of the latter on fuel nitrogen conversion to NO_x were determined by adding various levels of hydrogen sulfide to 1.0% ammonia-doped hot WGO. The results for both burners are shown in Figure 6. Hydrogen sulfide was metered in from 0.02 to 2.89 volume percent of fuel input. Oxygen in the flue was kept at 1.8%, corresponding to the 10% excess air level. As can be seen, small amounts of fuel-sulfur significantly enhanced fuel-nitrogen conversion to NO_x. Above about 1.5%, and up to 2.9%, hydrogen sulfide did not add greatly to further total NO_x enhancement. These results were essentially the same for both burners.

A review of the literature (5) indicated that the degree of initial fuel/air mixedness had an appreciable effect on fuel-nitrogen conversion to NO_x. With 1.0% ammonia-doped hot WGO at an excess air level held at 10% on the kiln burner, the amount of radial flow was varied from 0% to 36% of the total. The results, shown in Figure 7, confirmed the effect of initial mixing on fuel-NO_x emissions. The ammonia conversion to NO_x rose sharply from 10% radial flow to 36%. The thermal NO_x was not appreciably affected over the same range.

Following the single-dopant tests, so-called parametric studies were performed, wherein the various contaminants were added in combinations of two. The results of the ammonia-plus-tar and ammonia-plus-char parametric trials are presented in Table XI. The doping rates were representative of those used in the single-dopant tests. The total NO_x levels were found to be about 5% to 20% higher than expected on a simple additive basis (derived from the single-dopant studies), indicating some kind of synergistic effect.

The char-plus-tar parametric and the ammonia-plus-char-plus-tar ("dirty") doping results are shown in Table XII. Doping levels were again consistent with previous tests, and excess air was held at 10%. Total NO_x levels in the case of char-plus-tar appear to be purely additive while the results of the "dirty" doping trials show the same kind of NO_x enhancement as seen in the amonia-plus-char and ammonia-plus-tar tests. Total fuel-nitrogen conversion to NO_x is slightly higher on the kiln burner than on the baffle burner (∿10% versus ∿9%) if one assumes that thermal NO_x levels are not greatly affected by the sulfur in the dopants.

WELLMAN-GALUSHA AIR: EMISSION STUDIES

As with Wellman-Galusha oxygen (WGO) fuel gas, base-line data were obtained for clean Wellman-Galusha air (WGA) fuel gas at ambient and elevated temperatures on both the baffle and kiln burners. These data are presented in Table XIII along with the base-line natural gas data previously shown in Table VI. The fuels were fired at 1.03 ± 0.07 MW_t ($3.50 \pm 0.25 \times 10^6$ Btu/hr) with 10% excess air. For the baffle burner, natural gas and ambient (322 K) WGA were burned on the same 0.063-m ID (nominal 2-1/2 inch) fuel nozzle as WGO. Hot (616 K) WGA required a 0.078-m ID (nominal 3 inch Schedule 40) fuel nozzle to overcome the stability problems encountered on the 0.063-m ID nozzle. WGA fuel velocities were 63 m/s (206 ft/s) for ambient WGA and 83 m/s (273 ft/s) for hot WGA with combustion air at 19 and 30 m/s (62 and 99 ft/s), respectively.

On the kiln burner, the 0.108-m ID (nominal 4 inch) axial fuel nozzle was used for both ambient and hot WGA. The amount of radial flow was selected to give flame lengths comparable to that of natural gas; ambient WGA was fired with 10% radial flow, while hot WGA required 16% radial flow. Radial gas velocities were sonic while axial fuel velocities varied from 33 m/s (107 ft/s) for ambient WGA to 74 m/s (244 ft/s) for hot WGA. Air velocity was 3.1 m/s (10 ft/s).

Only "dirty" doped, hot WGA was studied. The results are shown in Table XIV. If the additives did not greatly affect thermal NO_x levels then the baffle burner is somewhat more efficient in converting fuel-nitrogen to NO_x than the kiln burner, namely, ~11% versus ~9%, respectively. This is opposite to the WGO results, where the kiln burner appeared to be slightly more efficient, in converting fuel nitrogen.

PARTICULATE STUDIES

Besides measuring the effects of the various dopants on gas-phase pollutants, total particulates in the stack were measured as well. The instrumentation and sampling technique used were described in a preceding section. Results of these total stack-particulate measurements are presented in Figure 8 for both WGO and WGA on the baffle and kiln burners. The fraction of char-derived particulates surviving in the stack increased from ~5% at a char input of 0.02 g/s to ~11% at a char input rate of 1.1 g/s. At all char doping rates, it was visually noted that a qualitatively large number of glowing particles were deposited on the furnace hearth. The presence of ammonia and/or tar did not affect the measured total stack-particulates from char.

In addition to total particulates, particle size distributions were measured for char-doped WGO on the baffle and kiln burners using a cascade impactor and methods described previously. The results are shown in Tables XV and XVI. On the baffle burner, with a char doping rate of 1.14 g/s, 63% of the particles are smaller than roughly 0.7μ, while on the kiln burner, for a char rate of 0.76 g/s, only about 5% of the particles are below 0.7μ. This shift in particle size depends more on char input level than on burner type for the cases here studied. This conclusion is supported by the data obtained from the total particulate measurements, wherein an increase in "coarse" particles trapped in the cyclone preseparator relative to "fine" particles staining the filter was observed with decreasing char doping rate for both burners.

FURNACE EFFICIENCY

Aside from the pollution aspects of burning low-Btu fuel gases as substitutes for natural gas, potential users of low-Btu gas are also very interested in retrofitting problems that might be encountered. One phase of any retrofit evaluation is the effect of firing low-Btu gas on furnace efficiency and heat transfer to the load. As noted in an earlier section, water-cooling tubes were positioned along the furnace wall to simulate the load of a steel reheat furnace for use with the baffle burner and then to simulate the calcining and reaction zones of a cement kiln for use with the kiln burner.

For the baffle burner, fired at 1.03 MW_t (3.5 X 10^6 Btu/hr), furnace thermal efficiencies, defined as the heat absorbed by the load divided by the fuel heat input, for clean ambient and clean WGO were found to be 30% and 31% as compared with the natural gas base-line value of 35%. Average flue temperatures were measured to be 1541 K (2314°F) for ambient WGO, 1564 K (2356°F) for hot WGO, and 1436 K (2126°F) for natural gas.

For clean ambient and clean hot WGA on the baffle burner, furnace thermal efficiencies were 24% and 29% with corresponding average flue temperatures of 1453 K (2156 °F) and 1504 K (2246°F).

With the kiln burner fired at 1.03 MW_t (3.5 X 10^6 Btu/hr), the base-line natural gas efficiency was 31% with an average flue temperature of 1532 K (2298°F). Clean ambient and clean hot WGO yielded efficiencies of 29% and 30% with average flue temperatures of 1524 K (2284°F) and 1545 K (2322°F).

On the kiln burner, clean ambient and clean hot WGA gave furnace thermal efficiencies of 24% and 28% with average flue-gas temperatures of 1479 K (2203°F) and 1553 K (2336°F).

In simulating the hot raw gasifier off-gas, the addition of dopants to the clean fuel might be expected to affect overall furnace thermal efficiency in two ways. First, the ammonia, char, and tar are fuels themselves and will therefore affect the fuel heat input. Second, char and tar could affect flame emissivity.

In the studies conducted, the level of contaminant doping was such that the maximum contribution to the total fuel heat input was less than 6%. Flue temperature measurements were essentially constant for a given fuel/burner with and without doping, indicating that the doping levels employed did not significantly affect fuel heat input. In any case, calculations of furnace efficiencies included the contributions of doped material to the total fuel enthalpy.

For ammonia additions of 1.0 volume percent, no effects on furnace thermal efficiency were observed for WGO or WGA on either burner. Char addition of 0.13 g/s (0.4 grains/SCF of WGO, 0.3 grains/SCF of WGA) also had no effect on efficiency for both fuels on the burners studied.

At a tar feed rate of 0.58 g/s (1.8 grains/SCF of WGO, 1.4 grains/SCF of WGA), furnace thermal efficiencies for WGO and WGA were increased by about 1.0% to 2.0% on both burners. This enhancement is probably due to the increase in flame luminosity that was visually observed.

DISCUSSION

FUEL-NITROGEN EFFECTS ON NO_x

In premixed flames, NO_x levels associated with thermal fixation of atmospheric nitrogen are dependent primarily on flame temperature and, secondarily, on the amount of combustion air (6). In turbulent diffusion flames, thermal NO_x has also been found to vary with the initial degree of fuel/air mixing (5). For the fuels and burners studied, base-line thermal NO_x levels, presented in Tables VI and XIII, are mainly ordered by adiabatic flame temperatures. Mixing effects on thermal NO_x associated with the different aerodynamic characteristics of the two burner types appear to be similar except for natural gas where a higher thermal NO_x level is found with the baffle burner than with the kiln burner, suggesting (after Reference 5) that natural gas/air mixing is somewhat better on the baffle burner.

Another source of NO_x in combustion is chemically bound nitrogen in the fuel. Since fuel-nitrogen bonds are much weaker than the bond in molecular nitrogen, fuel-nitrogen can give rise to higher amounts of NO_x than from thermal fixation (5,6). In a flame environment, fuel-nitrogen is generally believed to react through the competitive paths (6,4):

$$NH_i + Ox = NO \tag{1}$$

$$NH_i + NO = N_2 \tag{2}$$

where NH_i is some fuel-nitrogen intermediate, usually considered to be atomic nitrogen (7) or a cyano or amine derivative (8), and Ox is an oxygen-containing species such as O, OH, or O_2. In fuel-lean combustion, fuel-nitrogen appears in the exhaust gases mostly as NO and N_2; under fuel-rich conditions, significant HCN and NH_3 can also be found (8).

Factors that affect fuel-nitrogen conversion to NO_x are availability of oxygen, initial fuel-nitrogen concentration, temperature, and general fuel type (6,8). As the amount of oxygen available for combustion increases, the conversion of fuel-nitrogen to NO_x increases (9,6). Where fuel/air mixing is incomplete, this conversion is strongly affected by local oxygen concentrations as well as local flame temperatures (10). Combustion of fuel-nitrogen under locally fuel-rich conditions can lower the amount of NO_x formed (10) with the fuel-nitrogen preferentially forming N_2 (6). As initial fuel/air mixing is improved, the conversion efficiency of fuel-nitrogen to NO_x is enhanced (5). For example, one group (11) reported a doubling in fuel-NO_x when going from diffusional to premixed combustion.

With increasing levels of fuel-nitrogen, the fraction converted to NO_x decreases (11,12) even though the absolute amount increases. Unlike thermal NO_x, temperature does not greatly affect fuel NO_x in premixed flames (6), probably because overall fuel-nitrogen reactions are exothermic (5) and are therefore less temperature dependent.

Recent results (8) imply that fuel-nitrogen conversion to NO_x depends on fuel type; that is, ammonia conversion to NO_x in hydrocarbon combustion was found to be much greater than with a hydrogen/carbon monoxide fuel. The authors attributed this to a difference in the intermediate fuel-nitrogen species (NH_i in Equation 1), depending on whether the main fuel was a hydrocarbon or not (8).

In order to gauge the magnitude of several of the above parameters on fuel-nitrogen conversions to NO_x for raw gasifier effluents, char, tar, and ammonia were added to hot clean Wellman-Galusha fuel gases. To ascertain the contribution of each dopant to total NO_x emissions, single dopant and combination tests were also performed. The tests were performed using two industrial burners: a forward flow and a kiln burner.

With char-doped hot WGO on both burners, we have seen (Table VII) that total NO_x levels were not greatly increased above thermal at the char feed rates employed (maximum increase 30 ppm). The contribution of char-nitrogen (0.66 weight percent) to these increments is difficult to interpret, owing to the small change (if any) over thermal NO_x relative to measurement reproducibility ($\pm$ 5 ppm), and considering the known, but in this case unmeasurable, enhancement of thermal NO_x by fuel-sulfur (1.64 weight percent of char) in turbulent diffusion flames (9). Measured SO_2 at the flue entrance accounted for about 50% of the char-sulfur in all cases. Attempts to close this sulfur balance were unsuccessful.

As with char, hot WGO tar doping results, shown in Table VIII, cannot be unambiguously analyzed because of the presence of sulfur in the tar (0.47 weight percent). For the tar feed rates employed, measured NO_x exceeded the clean thermal values by about 30 ppm on both burners. This increase in NO_x

cannot be solely accounted for by tar-nitrogen (0.55 weight percent) even assuming a 100% conversion to NO_x. The implication is that tar-sulfur is enhancing thermal NO_x. The contributions to total NO_x of sulfur-enhanced thermal and fuel NO_x cannot be apportioned. If all tar-nitrogen were converted to NO_x, then sulfur enhancement of thermal NO_x could be up to 10 ppm over the clean, undoped value. Conversely, enhanced thermal NO_x could be much higher, with fuel NO_x contributing relatively little to the observed increase. Measured SO_2 accounted for only 20% of tar-sulfur in all cases, while the tar residue trapped in the fuel nozzles accounted for another 10% of the fuel-sulfur.

When fuel-sulfur is absent, as in the case of ammonia doped hot WGO, the effects of fuel-nitrogen on NO_x are more clearly evident. Varying the amount of ammonia in the fuel shows that the fraction of fuel-nitrogen converted to NO_x decreases with increasing fuel-nitrogen content, as can be seen from Figures 4 and 5, even though absolute NO_x levels increased. This observation is in agreement with the references cited at the beginning of this section. For example, on the baffle burner at 10% excess air, ammonia doped at 1.0 volume percent of fuel input yielded a 7% conversion to NO_x, while 0.4% ammonia gave a 14% conversion. With the kiln burner, the corresponding conversions were 8% and 16%.

Examination of Figures 4 and 5 shows that an increase in excess air, from 10% to 20%, enhanced fuel-nitrogen conversion to NO_x as expected. On the baffle burner with 1.0% ammonia in hot WGO, the conversion efficiency to NO_x increased from 7% to 8% with the increase in excess air. At the 0.4% ammonia level, the conversion went from 14% to 16%. Similar results were obtained with the kiln burner. For 1.0% and 0.4% ammonia, the respective increases in conversion with increased excess air were 8% to 10% and 16% to 17%.

At the radial flow chosen to give the proper flame length for the kiln burner (22% of the total hot WGO flow), fuel NO_x is only slightly higher than for the baffle burner at comparable ammonia doping rates and excess air. In other words, the anticipated mixing/aerodynamic effects of the different burner types were not evident at the operating conditions employed. However, changing the amount of radial flow drastically affected ammonia conversion to

NO_x, as can be seen from Figure 7. Increasing radial flow from 22% to 36% resulted in about a 50% increase in ammonia conversion, indicating that improved fuel/air mixing raises fuel-nitrogen conversion, as expected from the brief literature survey presented earlier (5). (This effect was also confirmed when fuel-nitrogen conversion was found to be 50% higher on a highly mixed high-forward-momentum burner than on the kiln and baffle burners in other tests done at IGT.) Raising the radial flow from 0% to 15% lowered fuel NO_x. In this region, the apparent loss and then recovery of fuel/air mixedness is probably due to a trade-off between increasing radial mixing and decreasing axial fuel momentum, the net effect of which is to decrease overall mixing up to about 10% radial, where radial flow becomes the dominant mixing parameter due to the radial flow penetration of the axial flow.

Since the ammonia-to-NO_x conversion is nearly the same for the baffle burner and the kiln burner (22% radial) with the same doping rate, it may be inferred that the baffle burner gave about the same degree of initial hot WGO/air mixing as the kiln burner at 22% radial flow.

FUEL-SULFUR EFFECTS ON NO_x

As noted in the discussion of the char and tar results, fuel-sulfur is known to affect thermal NO_x. In turbulent diffusion flames, characterized by relatively poor initial fuel/air mixing, fuel-sulfur enhances thermal NO_x while in premixed flames an inhibition occurs (9). Besides thermal NO_x, fuel-sulfur also affects fuel NO_x. In premixed flames it may enhance, inhibit, or have no effect on fuel NO_x depending on the point of sampling and/or the burning mixture's residence time in the combustion apparatus, while in turbulent diffusion flames, fuel-sulfur has been found to enhance fuel NO_x (9).

In order to determine the effects of various levels of fuel-sulfur (hydrogen sulfide) on fuel-nitrogen conversion to NO_x, hot WGO, doped with 1.0 volume percent ammonia, was fired on both burners with the results shown in Figure 6. Neglecting fuel-sulfur/thermal NO_x interactions, the anticipated enhancement of fuel-nitrogen conversion during turbulent diffusion combustion is evident at hydrogen sulfide levels of 0.5 to 2.9 volume percent fuel input. Fuel-sulfur effects on fuel NO_x are essentially the same for both burners. This is not surprising, because the kiln burner, operated at 22% radial flow, appears to give the same degree of initial hot WGO/air mixing as the baffle burner. This was also implied by the ammonia doping tests as previously noted. Further, as fuel-sulfur levels are increased, the enhancement of NO_x appears to reach a maximum, as suggested by Figure 6.

On both burners, measured SO_2 corresponds to about 80% of the sulfur input (as hydrogen sulfide) at all doping rates. The fate of the remaining sulfur is uncertain. If it were present as some other species, a possible candidate is SO_3. Although equilibrium considerations predict negligible amounts of SO_3 (13) in hot ($T \geq 1300$ K) combustion gases, relatively high concentrations of SO_3 are possible under combustion conditions, where rapid cooling of combustion gases can "freeze" SO_3 (13,14) at superequilibrium values. Even so, reported SO_3 levels are usually only a few percent (15), though levels as high as 10% have been recorded (16). Such high levels are possible where quenching of SO_3 takes place by rapid cooling or by short residence time in the combustion chamber.

With hot WGO fired on the baffle and kiln burners, combustion takes place by turbulent diffusion, resulting in wide variations in local species concentrations and temperatures. The SO_2, readily formed from the added H_2S (17), might form SO_3 in two ways: a) by reaction with O atoms in high temperature, fuel-lean regions; and b) in lower temperature regions ($T \leq 1000$ K) where the right-hand-side of the equilibrium process, $SO_2 + 1/2\ O_2 = SO_3$, is not negligibly small (13). Although the SO_3/SO_2 approach to equilibrium is slow at lower temperatures (15), the presence of NO_x can catalyze the formation of SO_3 by (15,18) —

$$NO + 1/2\ O_2 \rightarrow NO_2 \qquad (3)$$

$$NO_2 + SO_2 \rightarrow SO_3 + NO \qquad (4)$$

For the tests performed with ammonia and hydrogen sulfide-doped hot WGO, high concentrations of NO_x were present, making more plausible the possibility of high SO_3. That SO_2 levels measured were not severely depressed by some artifact of the sampling system/instrumentation is supported by subsequent IGT tests performed on a high-forward-momentum burner where measured SO_2 accounted for 95% of the hydrogen sulfide added to a 1.0 volume percent ammonia-doped low-Btu fuel (ambient WGO plus 25% N_2). Because the mixing characteristics of this kind of burner are superior to the baffle or kiln (22% radial) burners, one would expect less low-temperature formation of SO_3 if Reaction 4 is of any importance. The results are in good qualitative agreement with this tentative mechanism, though more research on this possibility is required before any definite conclusions can be made.

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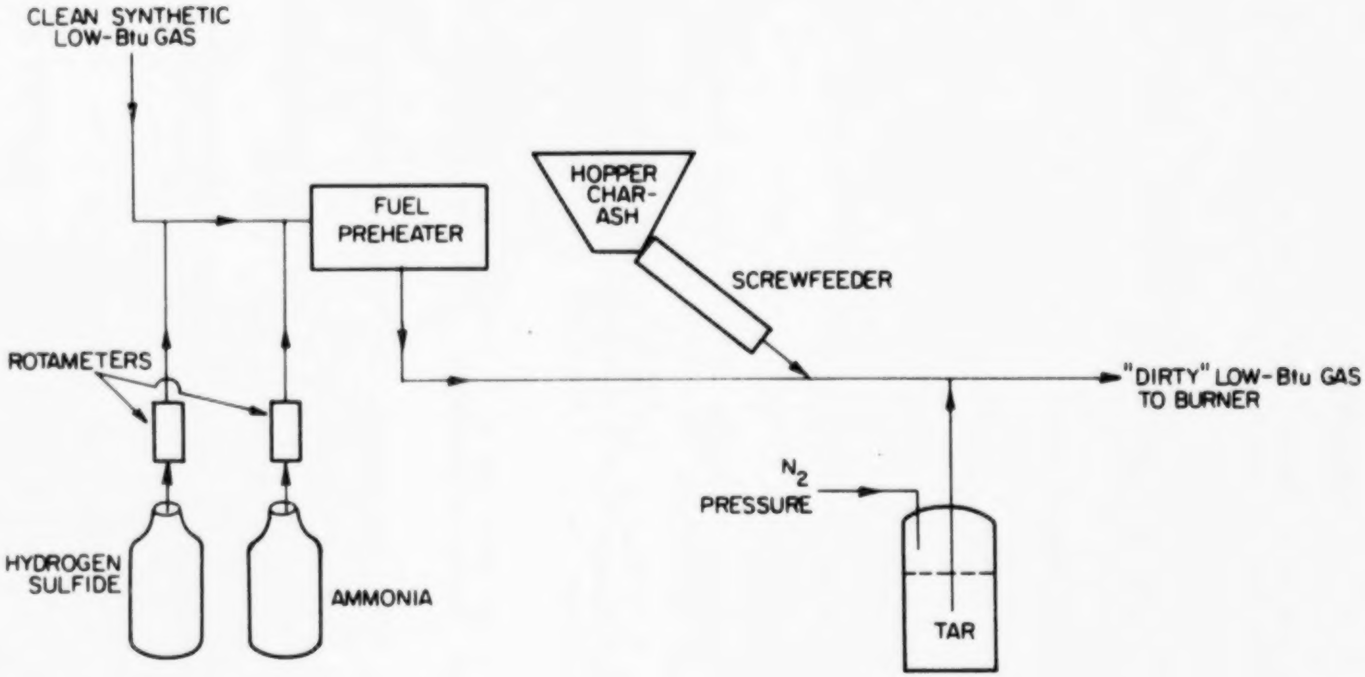
CLEAN SYNTHETIC LOW-Btu GAS
FUEL PREHEATER
HOPPER CHAR-ASH
SCREWFEEDER
ROTAMETERS
"DIRTY" LOW-Btu GAS TO BURNER
N_2 PRESSURE
HYDROGEN SULFIDE
AMMONIA
TAR

Figure 1. Doping System to Synthesize "Dirty" Low-Btu Gases

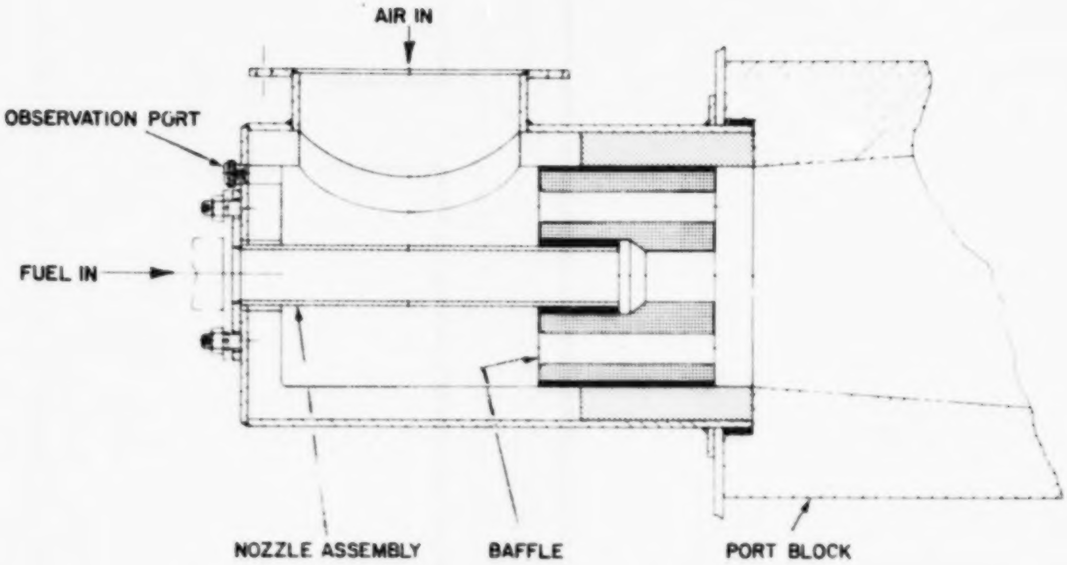
AIR IN
OBSERVATION PORT
FUEL IN
NOZZLE ASSEMBLY
BAFFLE
PORT BLOCK

Figure 2. Assembly Drawing of Baffle Burner

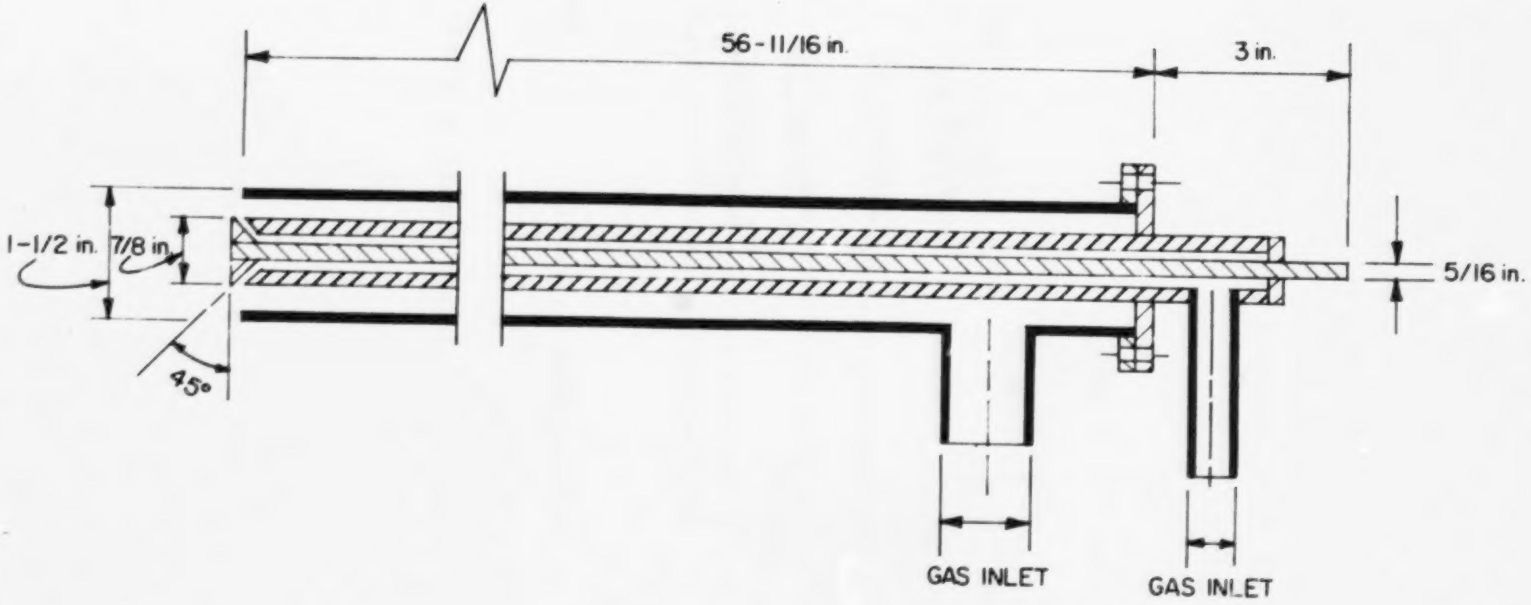
56-11/16 in.
3 in.
1-1/2 in.
7/8 in.
5/16 in.
45°
GAS INLET
GAS INLET

Figure 3. Schematic Diagram of the Kiln Burner Fuel Injector

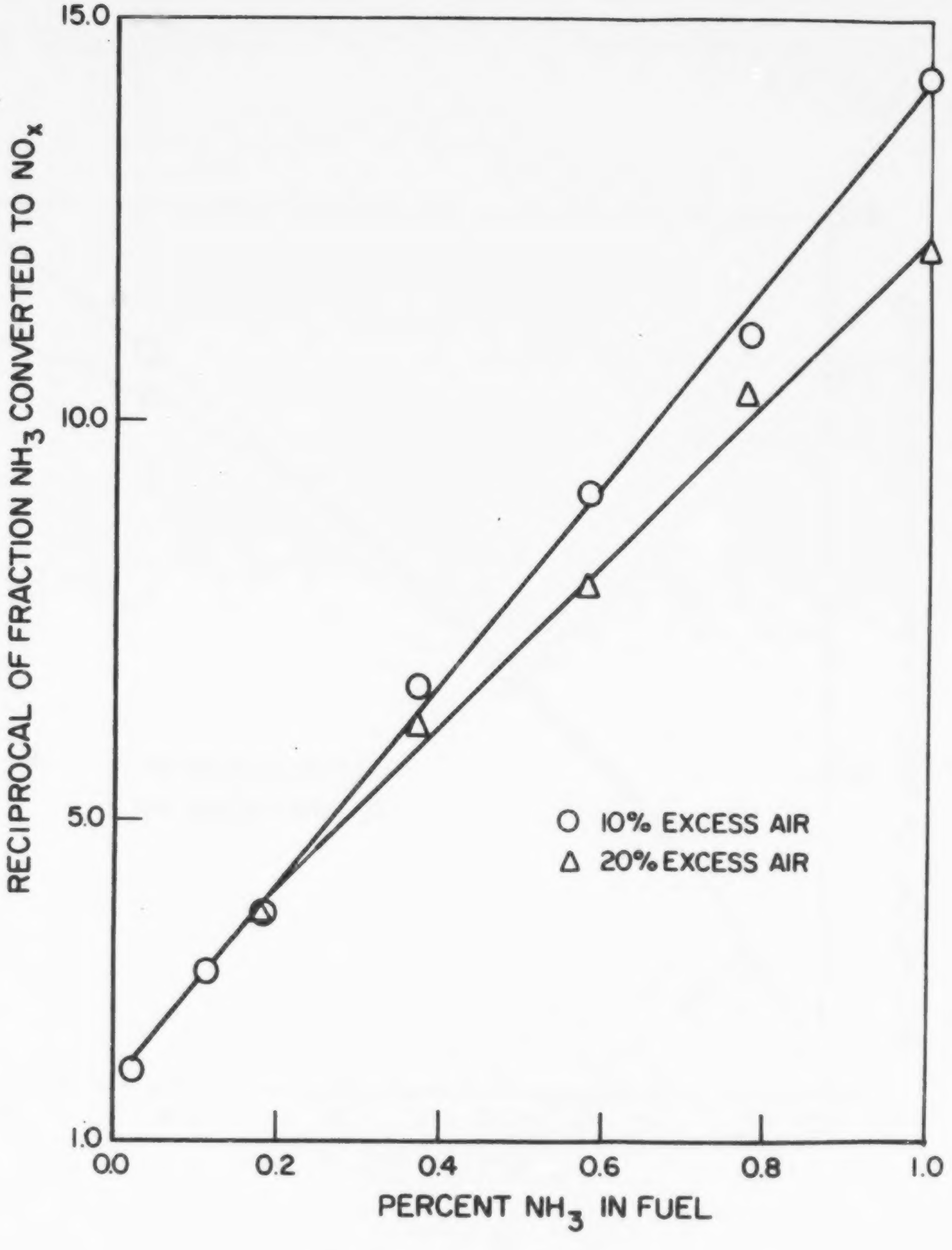

Figure 4. Ammonia Conversion for Wellman-Galusha Oxygen Fuel Gas on the Baffle Burner with 10% and 20% Excess Air

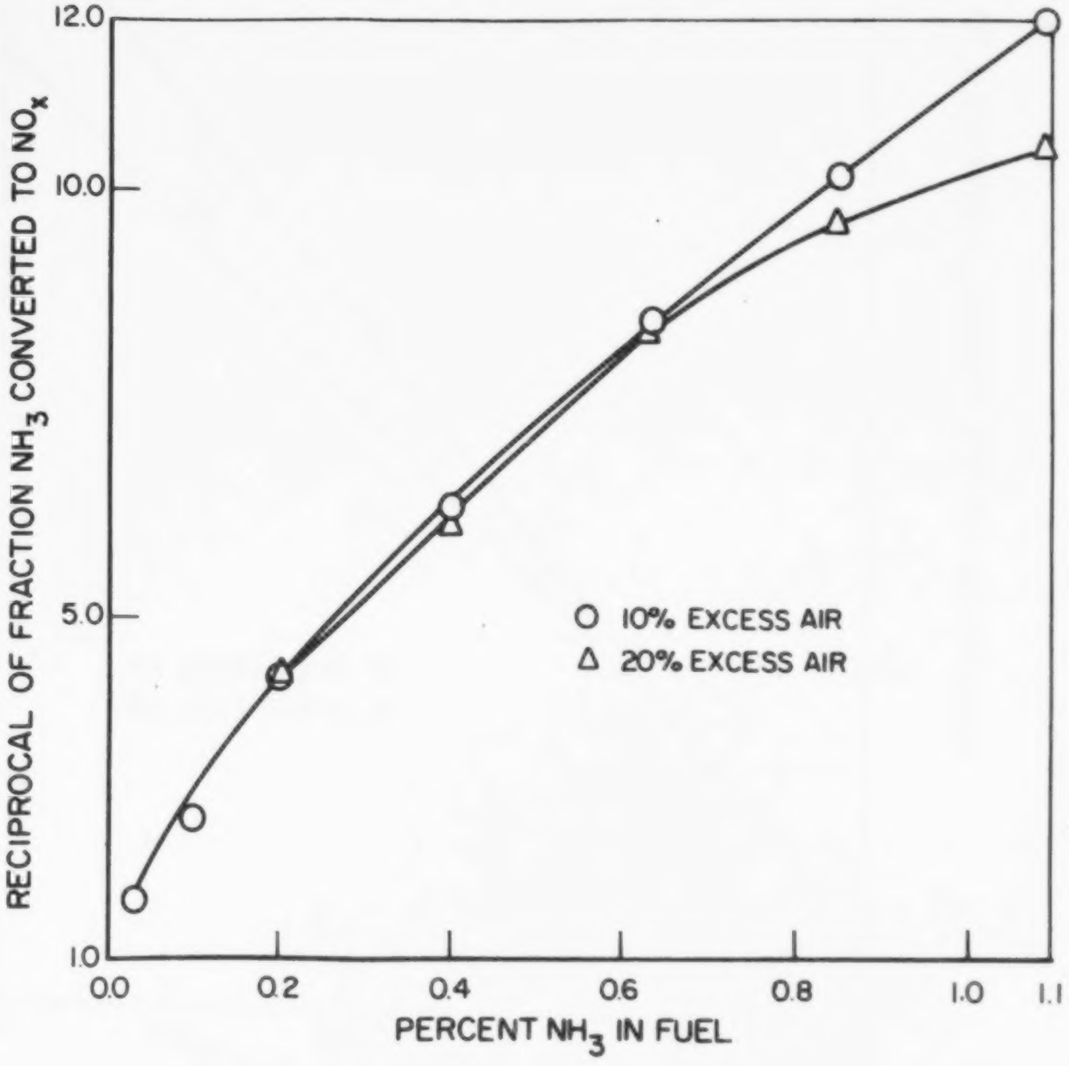

Figure 5. Ammonia Conversion for Wellman-Galusha Oxygen Fuel Gas on the Kiln Burner with 10% and 20% Excess Air

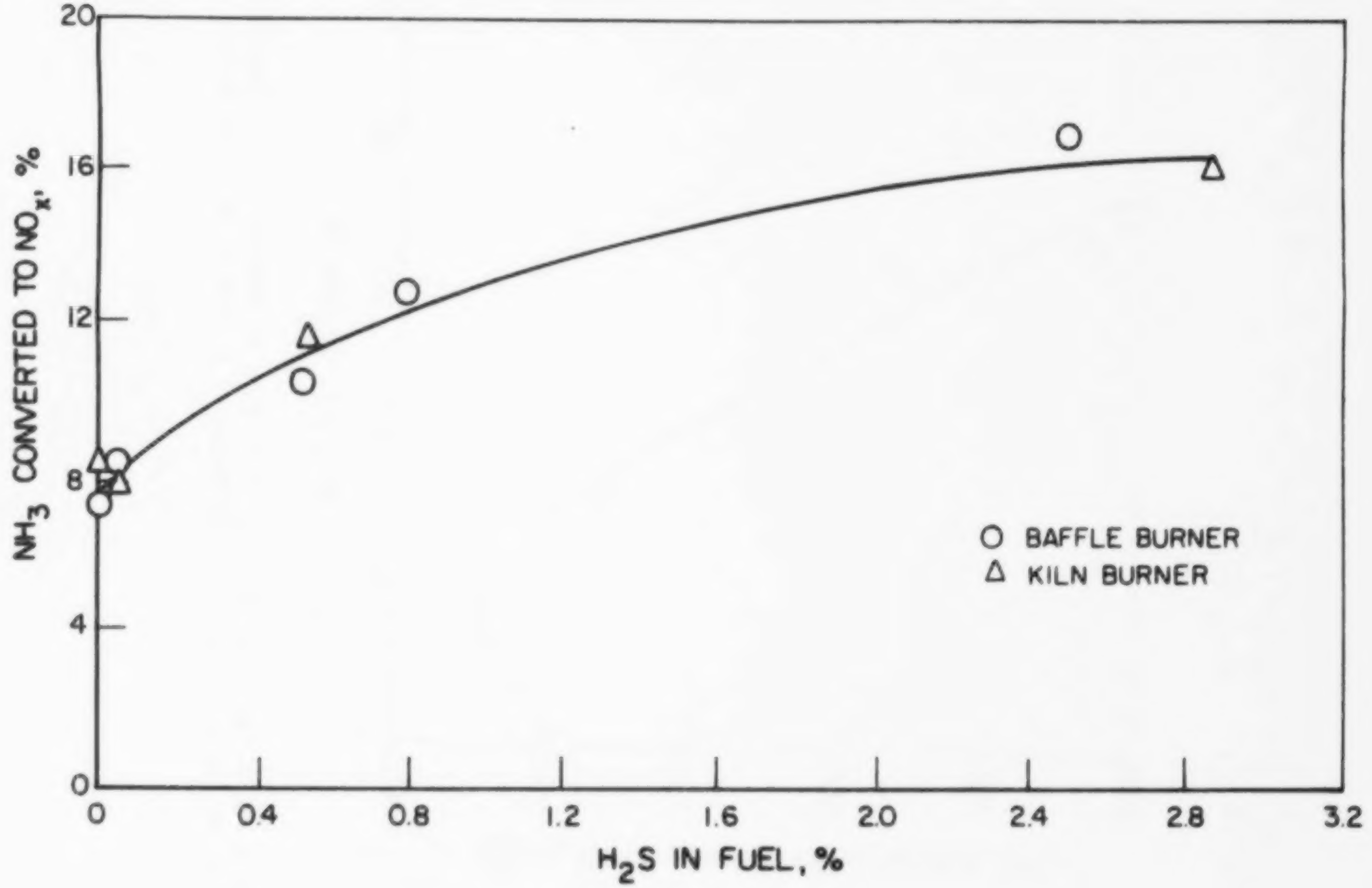

Figure 6. Effect of H_2S on Ammonia (1.0% by Volume) Conversion to NO_x with Wellman-Galusha Oxygen (10% Excess Air)

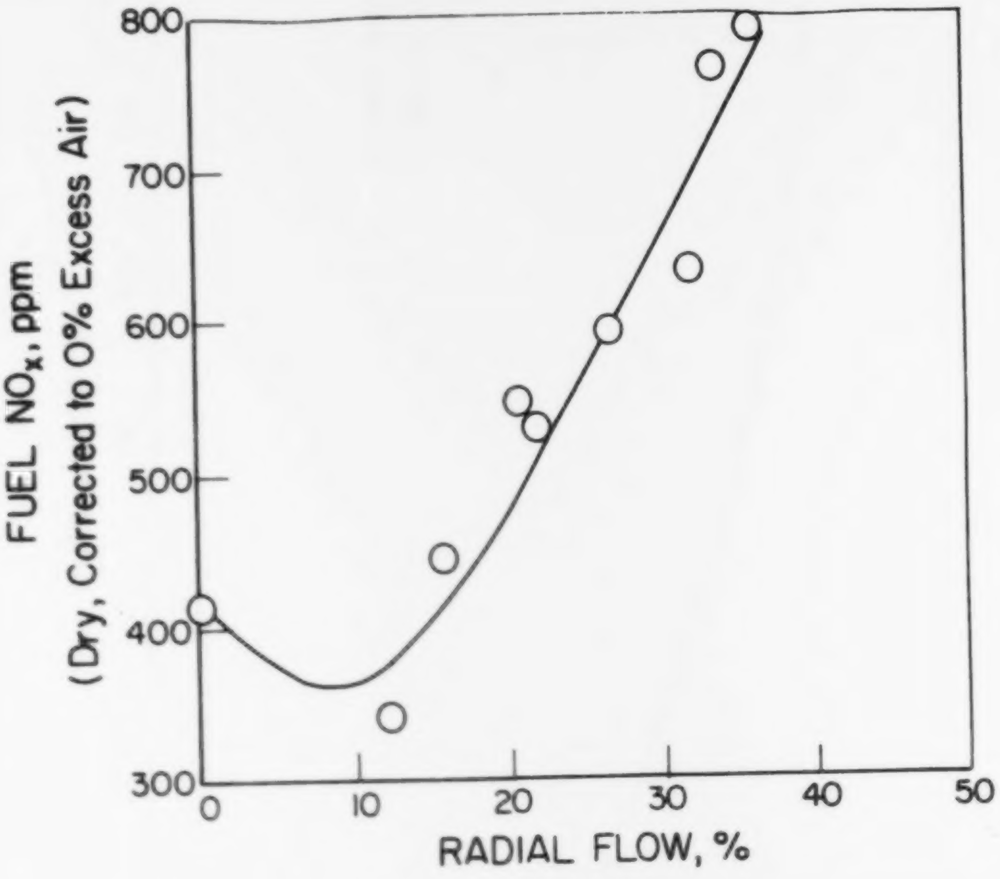

Figure 7. Effects of Radial vs. Axial Flow on Ammonia (1.0%) Conversion to NO_x with Wellman-Galusha Oxygen Fuel Gas on the Kiln Burner

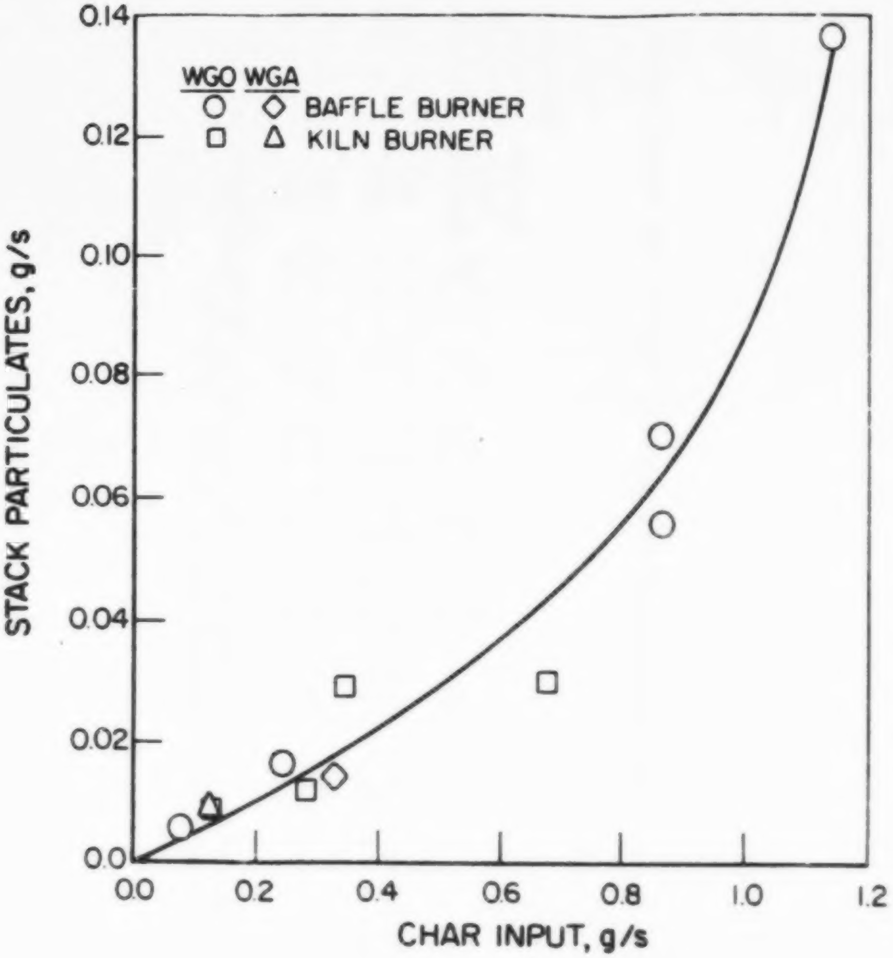

Figure 8. Particulate Emissions for Char Doping of Wellman-Galusha Fuel Gases on the Baffle and Kiln Burners (10% Excess Air)

TABLE I. FUEL COMPOSITION FOR LOW- AND MEDIUM- Btu GASES TESTED

Fuel	Temp, °F	CO	H_2	CO_2	CH_4	N_2	H_2O	Heating Value, Btu/SCF	Adiabatic Flame Temp,* K (°F)
Wellman-Galusha Oxygen	90	39.2	40.4	16.2	0.9	1.4	1.9	267	2248 (3587)
	800	29.7	30.6	12.4	0.7	1.1	25.5	202	2190 (3483)
Wellman-Galusha Air	90	26.9	14.3	7.4	2.6	46.9	1.9	159	2045 (3222)
	650	25.2	13.4	6.9	2.4	44.1	8.0	149	2044 (3220)

* 10% excess air at 477 K (400°F). The adiabatic flame temperature for ambient (298K, 77°F) natural gas is 2231 K (3356°F).

TABLE II. TAR ANALYSIS

Ultimate Analysis	wt % (Dry Basis)
Ash	0.0
Carbon	84.39
Hydrogen	5.65
Sulfur	0.47
Nitrogen	0.55
Oxygen (By Difference)	8.94

General Analysis	wt % (As Received)
Solids	27.1
Heavy Fraction	55.3
Light Fraction (Toluene, Benzene, Xylene)	17.6

TABLE III. CHAR ANALYSIS

Ultimate Analysis	wt % (Dry Basis)
Ash	22.88
Carbon	66.30
Hydrogen	1.75
Sulfur	1.78
Nitrogen	0.72
Oxygen (By Difference)	6.57

Proximate Analysis	wt % (As Received)
Moisture	7.8
Volatile Matter	13.6
Ash	21.1
Fixed Carbon	57.5

Sieve Analysis	wt % Retained
Screen	
200	67.7
230	3.3
270	4.9
325	3.6
Pan	20.5

TABLE IV. SUMMARY OF TEST DATA FOR BAFFLE BURNER

	Inputs					Flue-Gas Composition (as measured)							Particulate,	
	Rate	NH_3	H_2S	Tar	Char	NO	NO_2	CO	CO_2	O_2	SO_2	HC	g/m^3	g/s
Fuel Type	m^3/s	— % —		— g/s —		—— ppm ——			— % —		— ppm —			
Natural Gas	0.027	0	0	0	0	112	10	10	10.9	1.9	--	--		
WGO* (322K, 10%) Excess Air	0.111	0	0	0	0	38	10	200	24.0	1.8	--	--		
WGO* (700K, 10%) Excess Air	0.140	0	0	0	0	30	5	115	24.0	1.8	--	--		
	0.134	0.19	0	0	0	310	10	55	24.0	1.9	--	--		
	0.134	0.38	0	0	0	360	10	55	24.0	1.8	--	--		
	0.134	0.60	0	0	0	400	15	53	24.0	1.9	--	--		
	0.134	0.81	0	0	0	420	20	55	24.0	1.9	--	--		
	0.134	1.03	0	0	0	430	20	60	24.0	1.9	--	--		
WGO* (700K, 20%) Excess Air	0.134	0	0	0	0	38	5	45	22.0	3.3	--	--		
	0.134	0.19	0	0	0	290	10	45	22.0	3.3	--	--		
	0.134	0.38	0	0	0	355	10	45	22.0	3.2	--	--		
	0.134	0.60	0	0	0	410	20	40	22.0	3.2	--	--		
	0.134	0.81	0	0	0	435	20	40	22.0	3.2	--	--		
	0.134	1.03	0	0	0	465	20	40	22.0	3.3	--	--		
WGO* (700K, 10%) Excess Air Tar Doping	0.141	0	0	0.36	0	52	7	108	24.0	1.8	1	< 1		
	0.141	0	0	0.50	0	55	5	93	24.0	1.9	1	< 1		
	0.141	0	0	0.71	0	58	4	93	24.0	1.9	2	< 1		
WGO* (700K, 10%) Char Doping	0.138	0	0	0	0.13	30	3	59	24.0	1.8	4	< 1	0.037	0.009
	0.138	0	0	0	0.24	33	2	100	24.0	1.7	6	< 1	0.070	0.017
	0.138	0	0	0	0.86	44	4	90	24.0	1.9	29	< 1	0.263	0.064
WGO* (700K, 10%) Ammonia/Tar	0.139	0.99	0	0.39	0	520	10	108	24.0	1.7	2	< 1		
WGO* (700K, 10%) Ammonia/Char	0.138	1.00	0	0	0.13	470	43	65	24.0	1.8	4	< 1		
WGO* (700K, 10%) Char/Tar	0.138	0	0	0.46	0.13	40	8	63	24.0	1.9	7	< 1		
WGO* (700K, 10%) Ammonia/Char/Tar	0.138	1.00	0	0.47	0.13	500	42	75	24.0	1.8	8	< 1		
	0.136	1.02	0	0.38	0.07	510	15	24	24.0	1.9	7	< 1	0.026	0.006
WGO* + 15% FGR (700K, 10%) Ammonia/Char/Tar	0.138	1.00	0	1.22	0.13	510	20	45	24.0	1.9	4	< 1		
WGO* (700K, 10%) Hydrogen Sulfide	0.134	1.04	0.02	0	0	480	29	40	24.0	1.8	83	--		
	0.134	1.04	0.52	0	0	655	33	50	24.0	1.8	2546	--		
	0.136	1.02	2.53	0	0	950	50	37	24.0	1.9	11146	--		
WGA+ (322K, 10% Excess Air)	0.175	0	0	0	0	16	1	260	18.4	1.3	--	--		
WGA‡ (616K, 10% Excess Air)	0.175	0	0	0	0	22	2	35	18.4	1.4	--	--		
WGA‡ (616K, 10%) Ammonia/Char/Tar	0.186	1.04	0	0.58	0.35	610	20	40	18.4	1.4	8	< 1	0.044	0.016

* Wellman-Galusha oxygen fuel gas.

+ Wellman-Galusha air fuel gas.

‡ Wellman-Galusha air fuel gas. Uses 3-inch nozzle. (All others used 2-1/2 inch nozzle.)

TABLE V. SUMMARY OF TEST DATA FOR KILN BURNER

Fuel Type	Rate[d]	Inputs				Flue-Gas Composition (as measured)							Particulates,	
	m^3/s	NH_3 %	H_2S %	Tar g/s	Char g/s	NO ppm	NO_2 ppm	CO ppm	CO_2 %	O_2 %	SO_2 ppm	HC ppm	g/m^3	g/s
Natural Gas[a]	0.027	0	0	0	0	70	5	150	10.2	2.1	--	--	--	--
WGO[a,b] (316 K, 10% Excess Air)	0.104	0	0	0	0	35	5	405	24.0	1.9	--	--	--	--
WGO (705 K, 10% Excess Air)	0.127	0	0	0	0	24	4	34	24.0	1.8	--	--	--	--
	0.127	0.20	0	0	0	270	20	30	24.0	1.8	--	--	--	--
	0.127	0.40	0	0	0	360	30	34	24.0	1.8	--	--	--	--
	0.127	0.63	0	0	0	420	30	32	24.0	1.8	--	--	--	--
	0.127	0.85	0	0	0	460	40	25	24.0	1.8	--	--	--	--
	0.127	1.09	0	0	0	500	40	34	24.0	1.8	--	--	--	--
WGO (705 K, 20% Excess Air)	0.127	0	0	0	0	40	5	25	22.0	3.3	--	--	--	--
	0.127	0.20	0	0	0	270	20	25	22.0	3.3	--	--	--	--
	0.127	0.40	0	0	0	370	30	25	22.0	3.3	--	--	--	--
	0.127	0.63	0	0	0	420	30	25	22.0	3.3	--	--	--	--
	0.127	0.85	0	0	0	480	40	25	22.0	3.3	--	--	--	--
	0.127	1.09	0	0	0	550	50	20	22.0	3.3	--	--	--	--
WGO (705 K, 10% Excess Air) Hydrogen Sulfide	0.127	1.09	0.02	0	0	470	30	34	24.0	1.7	90		--	--
	0.127	1.09	0.54	0	0	700	40	34	24.0	1.7	2579		--	--
	0.127	1.09	2.89	0	0	920	80	34	24.0	1.7	12800	--	--	--
WGO (705 K, 10%) Char Doping	0.124	0	0	0	0.13	30	5	30	24.0	1.8	3	<1	0.037	0.009
	0.127	0	0	0	0.29	35	2	40	24.0	1.9	6	<1	0.050	0.012
	0.134	0	0	0	0.67	55	5	75	24.0	1.7	10	<1	0.129	0.031
WGO (705 K, 10%) Tar Doping	0.136	0	0	0.38	0	50	5	50	24.0	1.8	1	<1	--	--
	0.136	0	0	0.60	0	52	3	57	24.0	1.8	1	<1	--	--
	0.136	0	0	0.73	0	58	4	70	24.0	1.7	2	<1	--	--
WGO (705 K, 10%) Ammonia/Tar	0.129	1.07	0	0.63	0	540	40	75	24.0	1.9	1	<1	--	--
WGO (705 K, 10%) Ammonia/Char	0.144	0.98	0	0	0.13	480	30	33	24.0	1.8	3	<1	--	--
WGO (705 K, 10%) Char/Tar	0.136	0	0	0.48	0.35	65	5	45	24.0	1.9	6	<1	--	--
WGO (705 K, 10%) Ammonia/Char/Tar	0.136	1.02	0	0.25	0.35	540	40	40	24.0	1.8	7	<1	0.065	0.016
WGO + 15% FGR (705 K, 10%) Ammonia/Char/Tar	0.136	1.02	0	0.57	0.13	525	30	50	24.0	1.8	4	<1	--	--
WGA[c] (322 K, 10%) Clean	0.164	0	0	0	0	22	4	200	18.0	1.3	--	--	--	--
WGA (620 K, 10%) Hot/Clean	0.177	0	0	0	0	19	2	30	18.0	1.3	--	--	--	--
WGA (620 K, 10%) Ammonia/Char/Tar	0.172	0.99	0	0.45	0.13	460	30	35	18.0	1.3	7	<1	0.025	0.010

[a] Used 1-1/2 inch Schedule 10 (axial) fuel injector; all others used 4-inch Schedule '0 injector.

[b] Wellman-Galusha oxygen fuel gas.

[c] Wellman-Galusha air fuel gas.

[d] Fuel flow times heating value adjusted to give from 1.00 to 1.05 MJ/s input.

TABLE VI. BASE-LINE DATA FOR CLEAN FUELS:
NATURAL GAS AND WELLMAN-GALUSHA OXYGEN (WGO),
3.5 MILLION Btu/hr WITH 10% EXCESS AIR

	Fuel Type	Fuel Temperature, K	NO_x*	CO
			ppm	
Baffle Burner	Natural Gas	298	135	10
	WGO	322	53	200
	WGO	700	38	115
Kiln Burner	Natural Gas	298	83	150
	WGO	322	44	405
	WGO	700	31	34

* NO plus NO_2 (Dry, corrected to 0% excess air).

TABLE VII. CHAR DOPING: WELLMAN-GALUSHA OXYGEN (700 K), 3.5 MILLION Btu/hr WITH 10% EXCESS AIR

	Char Input, g/s	NO_x*	CO	SO_2
		ppm	ppm	ppm
Baffle Burner	0	38	115	0
	0.13	36	59	4
	0.24	38	100	6
	0.86	53	90	29
Kiln Burner	0	31	34	0
	0.13	38	30	3
	0.29	41	40	6
	0.67	66	75	10

* NO plus NO_2 (Dry, corrected to 0% excess air).

TABLE VIII. TAR DOPING: WELLMAN-GALUSHA OXYGEN (700 K), 3.5 MILLION Btu/hr WITH 10% EXCESS AIR

	Tar Input, g/s	NO_x*	CO	SO_2
		ppm	ppm	ppm
Baffle Burner	0	38	115	0
	0.36	65	108	1
	0.50	66	93	1
	0.71	68	93	2
Kiln Burner	0	31	34	0
	0.38	60	50	1
	0.60	60	57	1
	0.73	68	70	2

* NO plus NO_2 (Dry, corrected to 0% excess air).

TABLE IX. AMMONIA CONVERSION TO NO_x ON THE BAFFLE BURNER

% in Fuel	Excess Air, %	Fuel NO_x,* ppm	NH_3 Conversion, %
0.02	10	77	52
0.11	10	213	29
0.19	10	312	26
0.38	10	367	15
0.60	10	416	11
0.81	10	443	9
1.03	10	454	7
1.42	10	509	6
0.19	20	305	26
0.38	20	382	16
0.60	20	459	13
0.81	20	489	10
1.03	20	524	8

* Dry, corrected to 0% excess air.

TABLE X. AMMONIA CONVERSION TO NO_x ON THE KILN BURNER

% in Fuel	Excess Air %	Fuel NO_x,* ppm	NH_3 Conversion %
0.03	10	123	60
0.10	10	243	38
0.20	10	287	23
0.40	10	396	16
0.63	10	462	12
0.85	10	516	10
1.09	10	560	8
0.20	20	291	23
0.40	20	421	17
0.63	20	480	12
0.85	20	563	11
1.09	20	658	10

* Dry, corrected to 0% excess air.

Table XI. PARAMETRIC DOPING: WELLMAN-GALUSHA OXYGEN (700 K), 3.5 MILLION Btu/hr WITH 10% EXCESS AIR

	Inputs					
	NH_3, vol %	Tar g/s	Char g/s	NO_x* ppm	CO ppm	SO_2 ppm
Baffle Burner	0	0	0	38	115	0
	1.03	0	0	492	60	0
	0.99	0.39	0	580	108	2
	1.00	0	0.13	561	65	7
Kiln Burner	0	0	0	31	34	0
	1.09	0	0	591	34	0
	1.07	0.63	0	635	75	1
	0.98	0	0.13	558	33	3

* NO plus NO_2 (Dry, corrected to 0% excess air).

TABLE XII. PARAMETRIC AND "DIRTY" DOPING: WELLMAN-GALUSHA OXYGEN (700 K), 3.5 MILLION Btu/hr WITH 10% EXCESS AIR

	Inputs					
	NH_3, vol %	Tar g/s	Char g/s	NO_x* ppm	CO ppm	SO_2 ppm
Baffle Burner	0	0	0	38	115	0
	1.03	0	0	492	60	0
	0	0.46	0.13	53	63	7
	1.00	0.47	0.13	593	75	8
Kiln Burner	0	0	0	31	34	0
	1.09	0	0	591	34	0
	0	0.48	0.35	77	45	6
	1.02	0.25	0.35	635	40	7

* NO plus NO_2 (Dry, corrected to 0% excess air).

TABLE XIII. BASE-LINE DATA FOR CLEAN FUELS: NATURAL GAS AND WELLMAN-GALUSHA AIR (WGA), 3.5 MILLION Btu/hr WITH 10% EXCESS AIR

	Fuel Type	Temperature, K	NO_x* (ppm)	CO (ppm)
Baffle Burner	Natural Gas	298	135	10
	WGA	322	18	260
	WGA	616	26	35
Kiln Burner	Natural Gas	298	83	150
	WGA	322	28	200
	WGA	616	22	30

* NO plus NO_2 (Dry, corrected to 0% excess air).

TABLE XIV. "DIRTY" DOPING: WELLMAN-GALUSHA AIR (616 K), 3.5 MILLION Btu/hr WITH 10% EXCESS AIR

	Inputs					
	NH_3, %	Tar (g/s)	Char (g/s)	NO_x* (ppm)	CO (ppm)	SO_2 (ppm)
Baffle Burner	0	0	0	26	35	0
	1.04	0.58	0.35	672	40	8
Kiln Burner	0	0	0	22	30	0
	0.99	0.45	0.13	522	35	7

* NO plus NO_2 (Dry, corrected to 0% excess air).

TABLE XV. BAFFLE BURNER CASCADE IMPACTOR RESULTS FOR CHAR-DOPED WELLAMN-GALUSHA OXYGEN (700 K) — DOPING RATE: 1.14 g/s

Stage	Size (μ)	wt %	% Cumulative*
0	≥ 17.1	11.1	88.9
1	10.7 - 17.1	1.4	87.5
2	7.3 - 10.1	0.4	87.1
3	5.0 - 7.3	2.9	84.2
4	3.2 - 5.0	2.5	81.7
5	1.8 - 3.2	8.9	72.8
6	1.1 - 1.8	4.7	68.1
7	0.7 - 1.1	5.6	62.5
Filter	0 - 0.7	62.5	0

* Wt % of particulates passing through stage.

TABLE XVI. KILN BURNER CASCADE IMPACTOR RESULTS FOR CHAR-DOPED WELLMAN-GALUSHA OXYGEN (700 K)— DOPING RATE: 0.76 g/s

Stage	Size (μ)	wt %	% Cumulative*
0	≥ 12.6	48.5	51.5
1	7.9 - 12.6	4.9	46.6
2	5.3 - 7.9	9.2	37.4
3	3.6 - 5.3	7.2	30.2
4	2.4 - 3.6	5.8	24.4
5	1.2 - 2.4	12.1	12.3
6	0.7 - 1.2	7.8	4.5
7	0.5 - 0.7	0.2	4.3
Filter	0 - 0.5	4.3	0

* Wt % of particulates passing through stage.

SOME ASPECTS OF AFTERBURNER PERFORMANCE FOR CONTROL OF ORGANIC EMISSIONS

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SOME ASPECTS OF AFTERBURNER PERFORMANCE FOR EMISSIONS CONTROL

ABSTRACT

The initial phase of this program is intended to conduct an emission assessment of afterburner control systems based on available data. This phase would be a prelude to intended laboratory and field experimental efforts.

This paper reports on a portion of the Phase I environmental assessment. Firstly it reports on the use of existing data to estimate the potential national usage of afterburners, based on emissions. Secondly, it reports on an evaluation of field test data from the files of one local air pollution control agency. Results of the analyses show that in-service afterburners appear to be less efficient than are units reported on in much of the literature. The lower afterburners efficiency has little impact on national organic nonmethane emissions, but may have marked impact in local areas.

SOME ASPECTS OF AFTERBURNER PERFORMANCE FOR EMISSIONS CONTROL

INTRODUCTION

Organic emissions are of concern because of their participation in reactions leading to the production of oxidants and, because in certain locales, further reactions produce irritating smogs. One type of device that can be utilized for the control of hydrocarbon or organic emissions from some stationary sources is the afterburner or fume incinerator.

Afterburners can be applied to some, but not all organic emission sources. Generally they can be applied to organic emission sources having a well defined and contained emission stream, such as chemical, metallurgical, surface coating, and agricultural processes; they cannot readily be applied to sources having scattered and uncontained organic emissions, such as burning landfills and coal refuse piles, pipe leaks, and uncontained venting. Among the sources to which they are applied are:

- Resin kettels
- Varnish cookers
- Sulfuric acid manufacturing
- Phosphoric acid manufacturing
- Paint-bake ovens
- Wire-coating process
- Soap and synthetic detergent industries
- Glass manufacture
- Frit Smelters
- Food Processing Equipment
- Fish canneries
- Animal-matter rendering
- Electroplating
- Insecticide Manufacture
- Oil and solvent refining
- Chemical milling
- Coffee roasting
- Meat smokehouses
- Fertilizer plants
- Rotogravuring
- Degreasing operations
- Dry Cleaning
- Fiberboard drying and curing

With such widespread use of afterburners, it is important to national and local air quality efforts to understand just how successful afterburners are at controlling emissions. Unfortunately, the overall performance of afterburners is frequently based on limited research studies, or on tests of new units. These test measure afterburner performance with well-tuned units operating at peak efficiency. Consequently, efficiency values above 90 percent or 95 percent, and sometimes as high as even 99 percent, are usually reported. However, there remains the strong suspicion that performance of typical in-service afterburners is not as good as that reported in the idealized tests. Because afterburners are not income-producing devices, it is suspected that their maintainence, etc., receives less than adequate attention. This paper considers the extent of application of afterburners, and reports on afterburner performance based on results of a limited number of afterburner emission tests conducted by a local air pollution control agency.

APPLICATION OF AFTERBURNERS AS RELATED TO NATIONAL HYDROCARBON EMISSIONS

Based on the work of others(1) as summarized in Table 1, it is estimated that 106,548,900 metric tons of volatile organics are emitted in the U.S. each year from stationary (and natural) sources. Of this total, 25,212,400 MT/yr (metric tons per year) are volatile nonmethane organics. Further examining the total emissions of organics, these emissions may be divided between natural and man-related sources as follows:

Volatile organics	
Natural sources	85,300,000 MT/yr (80.1 percent)
Man-related sources	21,248,900 MT/yr (19.9 percent)
Volatile nonmethane organics	
Natural sources	9,100,000 MT/yr (36.1 percent)
Man-related sources	16,112,400 MT/yr (63.9 percent).

Limiting the analysis to volatile nonmethane organics, 62.2 percent of these emissions are from sources not considered amenable to the application of air pollution control devices (e.g. natural sources, fossil fuel extraction, open burning). The remainder, about 37.8 percent of the volatile nonmethane organic emissions, originate from processes amenable to the application of emission control devices. Figure 1 graphically illustrates the above distribution of volatile nonmethane organics between natural sources and between man-related sources for which emission control devices are considered applicable or not applicable.

Further considering the volatile nonmethane organic emissions from sources for which control devices are applicable, Reference 1 estimates that about 75.7 percent of these emissions (28.6 percent of total volatile nonmethane organic emissions) could be controlled with the application of control devices. The remaining 24.3 percent of such emissions are estimated to be emitted due to control devices being less than 100 percent efficient. Figure 1 shows these data graphically.

Based on data in Reference 1, afterburners are potentially applicable to sources representing 18.3 percent of all volatile nonmethane organic emissions. These sources emit 4,618,500 MT/yr of such emissions. By assuming an afterburner efficiency of 90 percent for nearly all afterburner applications, the authors of Reference 1 calculated a control efficiency of 87.9 percent for all sources amenable to the use of afterburners. Thus, for sources for which afterburners are applicable, the controlled volatile nonmethane organic emissions are 4,057,800 MT/yr and the uncontrolled emissions are 560,700 MT/yr. Figure 2 repeats the data shown on Figure 1 but includes a further breakdown of emissions from controllable sources into sources that are controllable with afterburners and sources that are not controllable with afterburners.

It should be recognized, however, that afterburners would not be used on all sources to which they might be applied; other control devices might be selected due to lower cost or better compatibility with the process. Thus, if afterburners were applied to one-half of those sources amenable to the application of afterburners, the national figures for afterburner controlled sources would be:

inlet emissions	2,309,250 MT/yr
outlet emissions	280,350 MT/yr
controlled emissions	2,028,900 MT/yr

EVALUATION OF AFTERBURNER TEST RESULTS

An evaluation was made of the results of 73 field tests on afterburners conducted by a local air pollution control agency. These tests were conducted over a period of 14 years and do not appear to be limited to tests of newly installed units. In fact, a few of the tests were made as a result of citizen complaints regarding the emission source. Hence, these tests would appear to be more representative of in-service afterburner performance than are most other sources.

The results of these tests were revealing in the poor afterburner performance recorded for many of the tests. Figure 3 shows the distribution of afterburner efficiencies based on emissions of total nonmethane organic species. The median efficiency was about 76 percent. About 38 percent of the tests gave afterburner efficiencies of 90 percent or higher. Another 18 percent of the tests gave efficiencies of 70 to 90 percent, and 25 percent of the tests gave efficiencies from 0 to 70 percent. Finally, 19 percent of the tests recorded afterburner efficiencies below zero percent, that is, outlet emissions exceeded inlet emissions.

Figure 4 shows the distribution of afterburner efficiencies for the same tests but based on reactive organics. (Reactive organics include aromatics, phenols, carbonyls, and organic acids.) Afterburner efficiencies for reactive organics were lower than for total nonmethane organics; the median efficiency was only 50 percent. Twenty-nine percent of the tests gave efficiencies, based on reactive organics, or 90 percent or above; another 14 percent were in the 70 to 90 percent range; and 25 percent were in the 0 to 70 percent range. Finally, 33 percent of the tests gave negative efficiencies.

The negative efficiencies based on reactive organics is understandable in that, if reactive organics are only a small fraction of total organics at the inlet, an ineffective afterburner could convert a fraction of the nonreactive organics into reactive organic species and produce a negative efficiency. A negative efficiency based on total nonmethane organics is not so easily explained. Efficiencies that are only slightly negative (as a number were) might be explained by limits in the accuracy of organic sampling and analysis and/or of flow rate measurements. However, efficiencies that are negative by more than about 30 percent cannot readily be explained in this way. A few afterburners were operating at off-design conditions (1000 F to 1100 F conbustion zone temperatures versus 1300 F to 1500 F normally used); such units may have been producing nonmethane organics from the fuel gas due to poor combustion conditions.

Figure 5 is a cross tabulation of afterburner efficiencies for both total nonmethane organics and reactive organics. From Figure 5, it can be seen that 30 tests (56 percent) reported about the same efficiencies for both total and reactive organics. Also, 16 tests (29 percent) showed a higher efficiency for controlling total nonmethane organics and 8 tests (15 percent) showed a higher efficiency for controlling reactive organics. The large degree of scatter of data shown in Figure 5 shows that if, both total and reactive organics are considered important, it is necessary to test for each as the efficiency for controlling one type of emission may not permit estimation of control efficiency for the other type of emission.

IMPACT OF LESS EFFICIENT AFTERBURNERS

The impact of afterburners being less efficient than is generally assumed for enviornmental assessment studies has been evaluated, both on a national and a local basis.

Considering the national viewpoint, Table 1 and Figure 2 illustrates national emissions of volatile nonmethane organics from stationary sources with an assumed afterburner efficiency of 90 percent. National emissions of volatile nonmethane organics from sources utilizing afterburners was estimated to be 280,350 MT/yr. However, if afterburners are only 76 percent efficient (as shown earlier), instead of 90 percent efficiency, the outlet emission could be 596,000 MT/yr. This value is 315,650 MT/yr greater than the former value. Hence, based on the assumptions that:

1. the test data available are representative of the performance of all afterburners
2. afterburners are applied to 50 percent of the sources for which they are considered amenable,

volatile nonmethane organic emissions from afterburner controlled sources are presently underestimated by 315,650 MT/yr on a nationwide basis. This value is equal to about 1. 3 percent of total national volatile nonmethane organic emissions, and equal to about 2.0 percent of national man-related volatile nonmethane organic emissions. Such an error in estimating national emissions is probably not significant, as many of the values used in compiling Table 1 were not known to such accuracies.

Now, considering the impact on specific locales, it can reasonably be assumed that, for well populated areas, nearly all volatile nonmethane organic emissions are from sources amenable to controls. That is in populated areas uncontrolled sources such as open burning and solid waste disposal to other than incinerators would be absent and natural sources would be present in much smaller proportions then in the nation as a whole. Considering that afterburners are applied to one-half of the stationary sources in a populated area, the impact of poorer afterburners performance (76 percent versus 90 percent) would be an underestimation of volatile nonmethane organic emissions from stationary sources by up to 27 percent, depending on the distribution of types of industry. Such an error could have a significant impact on local air pollution control strategies, and in the evaluation of air pollution control efforts.

CONCLUSION

It has been shown that afterburners are candidate organic emission control devices for application to emission source emitting about 29 percent of the volatile nonmethane organic emission generated by man-related stationary sources. Further, the analysis of afterburner efficiency conducted as a part of this program shows that typical in-service afterburners are probably considerably less efficient then are the well tuned units used in the research studies normally reported in literature.

It has been shown that the error in estimating volatile nonmethane organic emissions due to the demonstrated poorer performance of afterburners is probably not significant of a national basis (causing an error of about 2 percent), but may be very significant on a local basis (causing an error that may exceed 27 percent). Hence, at least as it affects local areas, it is important that the effort to understand and interpret the actual performance of in-service afterburners be continued.

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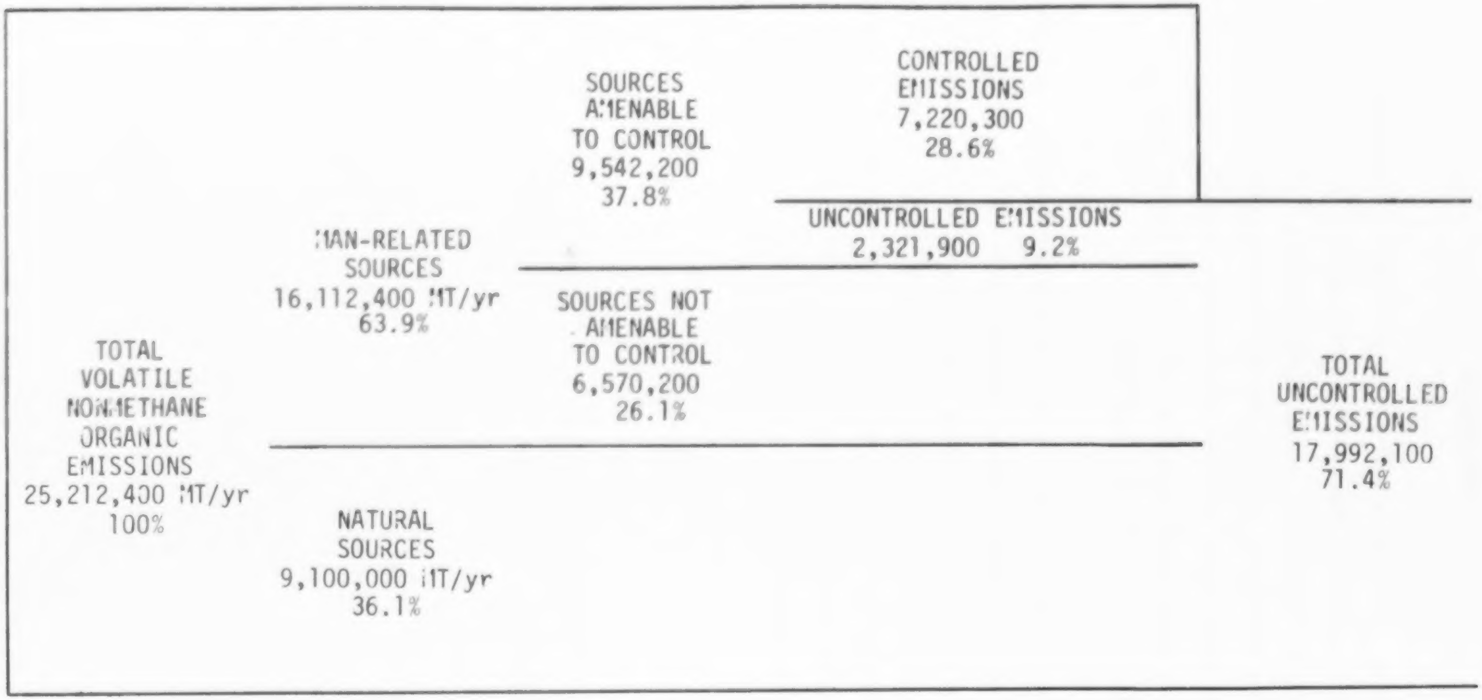
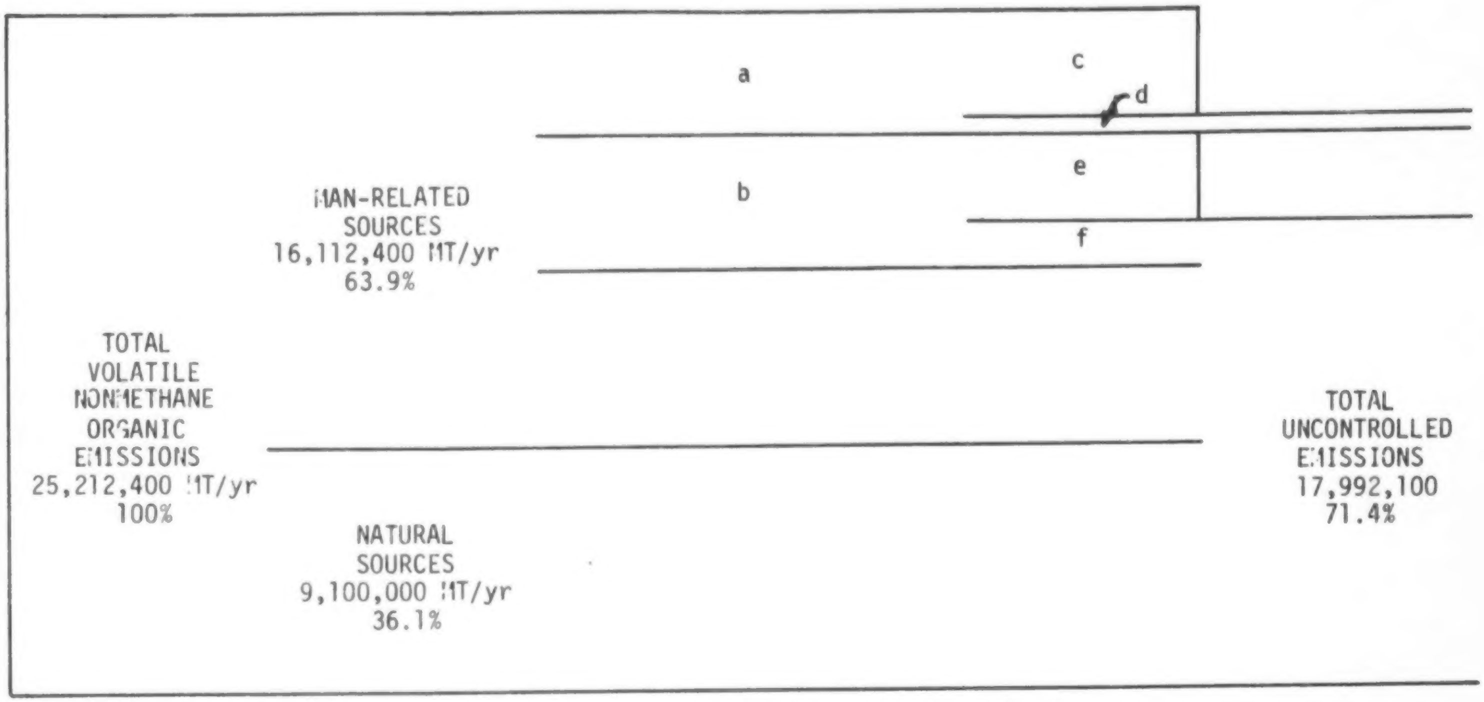

TOTAL
VOLATILE
NONMETHANE
ORGANIC
EMISSIONS
25,212,400 MT/yr
100%
MAN-RELATED
SOURCES
16,112,400 MT/yr
63.9%
NATURAL
SOURCES
9,100,000 MT/yr
36.1%
SOURCES
AMENABLE
TO CONTROL
9,542,200
37.8%
SOURCES NOT
AMENABLE
TO CONTROL
6,570,200
26.1%
CONTROLLED
EMISSIONS
7,220,300
28.6%
UNCONTROLLED EMISSIONS
2,321,900 9.2%
TOTAL
UNCONTROLLED
EMISSIONS
17,992,100
71.4%

FIGURE 1. NATIONAL VOLATILE NONMETHANE ORGANIC EMISSIONS FROM STATIONARY SOURCES


TOTAL
VOLATILE
NONMETHANE
ORGANIC
EMISSIONS
25,212,400 MT/yr
100%
MAN-RELATED
SOURCES
16,112,400 MT/yr
63.9%
NATURAL
SOURCES
9,100,000 MT/yr
36.1%
a
b
c
d
e
f
TOTAL
UNCONTROLLED
EMISSIONS
17,992,100
71.4%

a. SOURCES AMENABLE TO CONTROL WITH AFTERBURNERS; 4,618,500 MT/yr; 18.3%

b. SOURCES AMENABLE TO CONTROL BUT NOT WITH AFTERBURNERS; 4,923,700 MT/yr; 19.5%

c. CONTROLLED EMISSIONS FROM SOURCES AMENABLE TO CONTROL WITH AFTERBURNERS; 4,057,800 MT/yr; 16.1%

d. CONTROLLED EMISSIONS FROM SOURCES AMENABLE TO CONTROL BUT NOT WITH AFTERBURNERS; 3,162,500 MT/yr; 12.5%

e. UNCONTROLLED EMISSIONS FROM SOURCES AMENABLE TO CONTROL WITH AFTERBURNERS; 560,700 MT/yr; 2.2%

f. UNCONTROLLED EMISSIONS FROM SOURCES AMENABLE TO CONTROL BUT NOT WITH AFTERBURNERS, 1,761,200 MT/yr; 7.0%

FIGURE 2. NATIONAL VOLATILE NONMETHANE ORGANIC EMISSIONS FROM STATIONARY SOURCES SHOWING AFTERBURNER CONTROLLABLE EMISSIONS

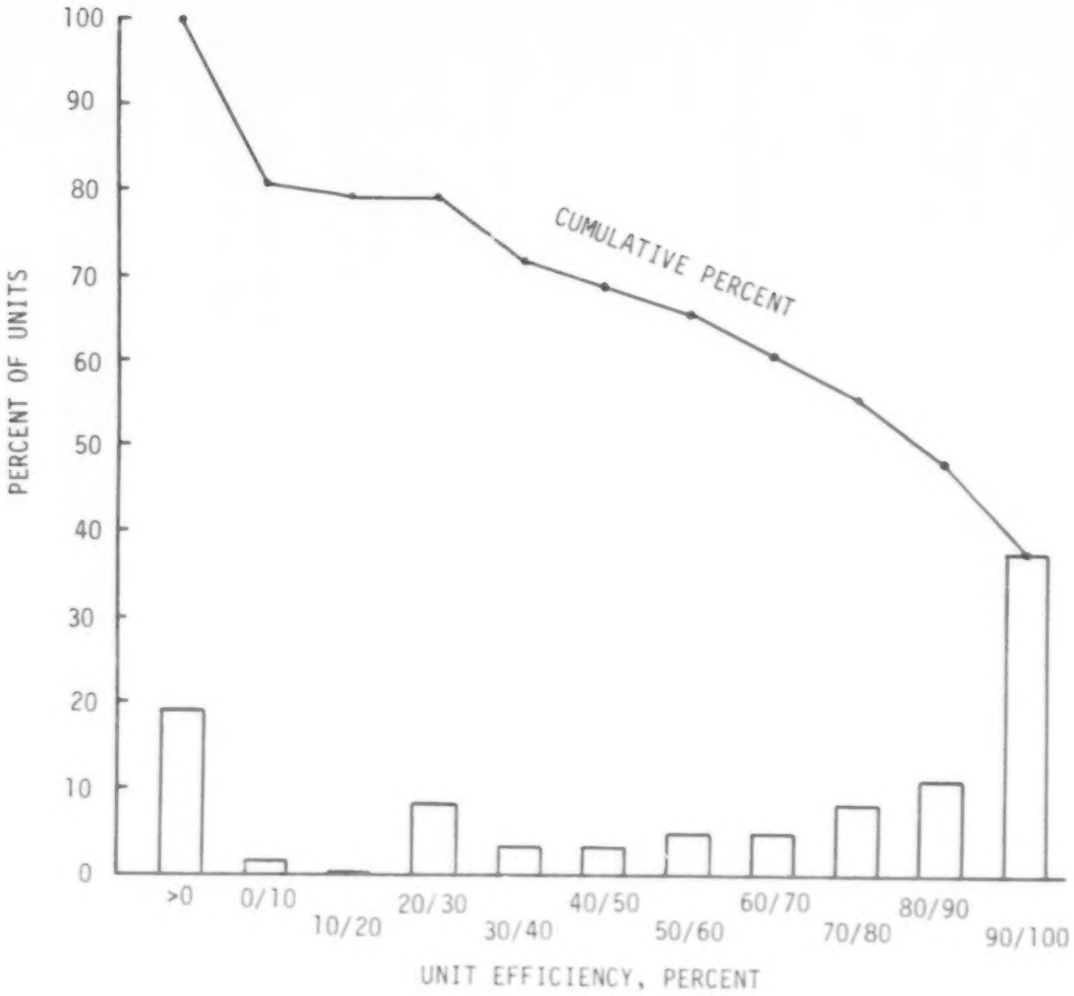


FIGURE 3. AFTERBURNER EFFICIENCY BASED ON TOTAL ORGANICS

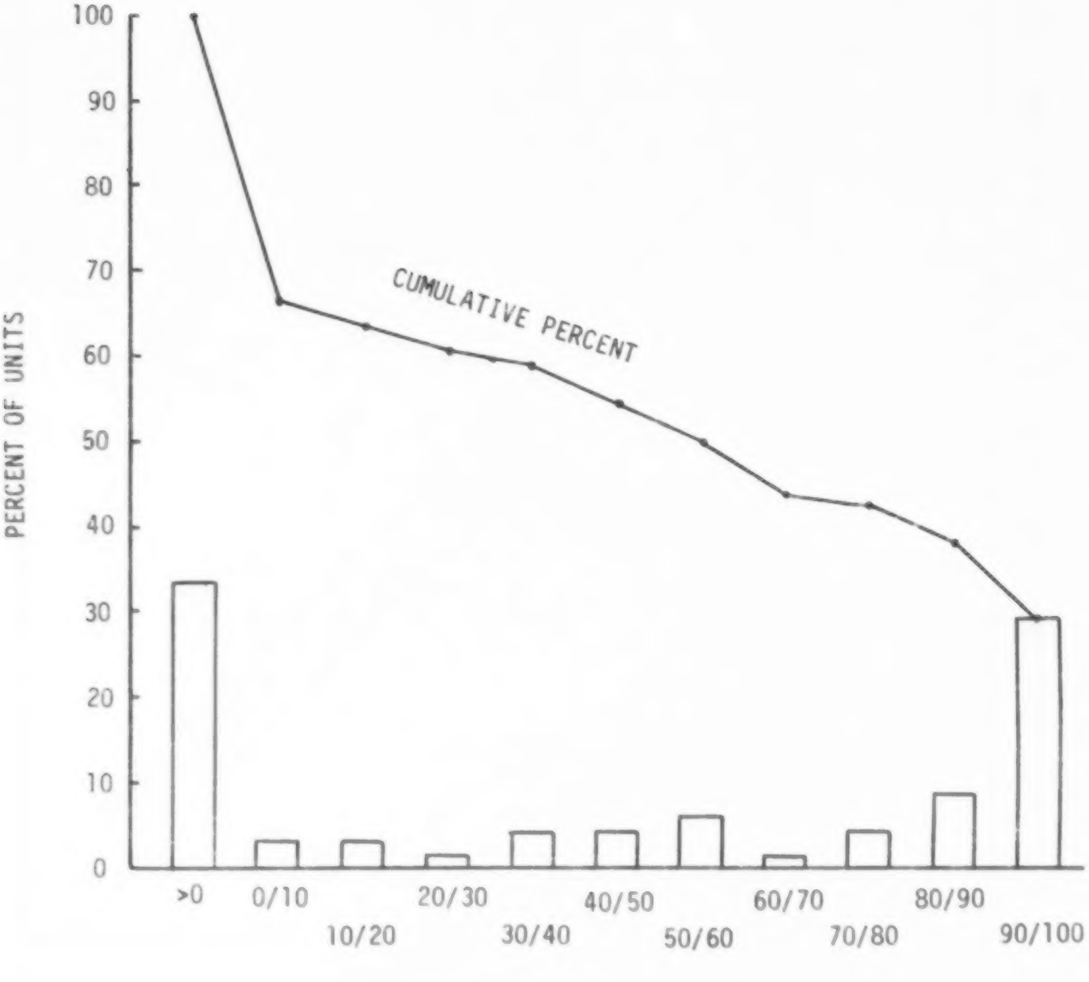


FIGURE 4. AFTERBURNER EFFICIENCY BASED ON REACTIVE ORGANICS

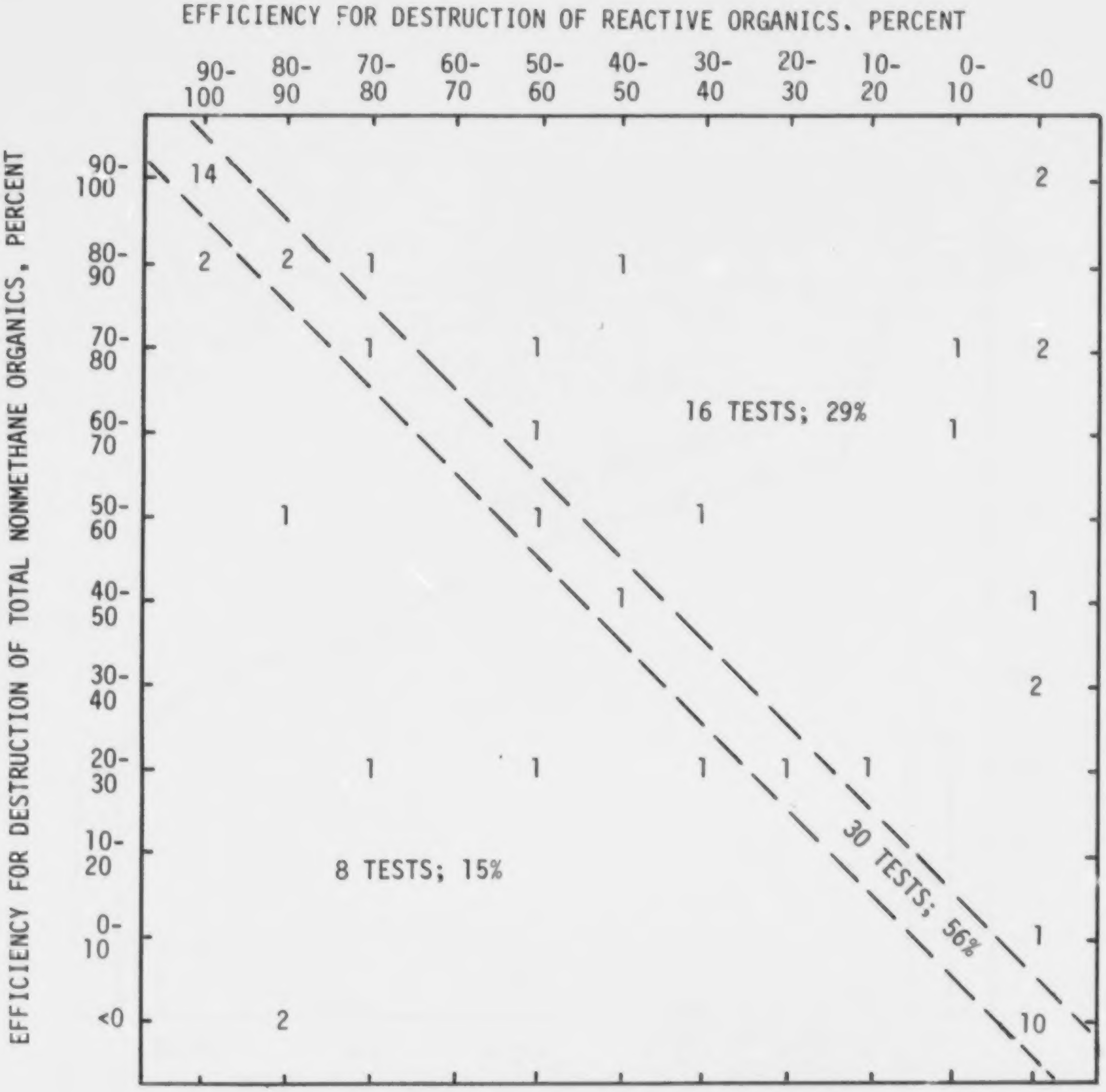


FIGURE 5. COMPARISON OF AFTERBURNER EFFICIENCIES FOR TOTAL AND REACTIVE ORGANICS

TABLE 1. SUMMARY OF NATIONAL ORGANIC EMISSIONS FROM STATIONARY SOURCES[a]

	Source Category	Total Atmospheric Emissions (MT/yr)			Uncontrolled Emissions from Processes Amenable to Air Pollution Controls (Uncontrolled Emissions from Processes Amenable to Control with Afterburners) (MT/yr)			Controllable Emissions from Sources Amenable to Controls (Controllable Emissions from Sources Amenable to Control with Afterburners) (MT/yr)		
		Volatile Organics	Volatile Nonmethane Organics	Organic Particulates	Volatile Organics	Volatile Nonmethane Organic	Organic Particulates	Volatile Organics	Volatile Nonmethane Organics	Organic Particulates
I	Fossil Fuel Extraction[b]	2,510,000	-	-	2,510,000	-	-	1,830,000	-	-
II	Fossil Fuel Processing[b]	1,716,400	-	7,300	1,714,000	-	7,300	1,030,000	-	3,650
III	Fossil Fuel Transportation Storage, & Distribution	2,071,000	2,071,000	77,300	2,071,000	2,071,000	77,300	1,363,000	1,363,000	69,600
IV	Fossil Fuel Refining	2,173,500	2,173,500	269,000	2,173,500 (117,000)	2,173,500 (117,000)	269,000 (0)	1,400,000 (116,000)	1,400,000 (116,000)	243,600 (0)
V	Fossil Fuel Combustion[c]	724,000	383,900	-	317,000	68,200	-	314,000	67,500	-
VI	Fossil Fuel Feedstock Chemical Processing	1,400,000	1,077,000	45,800	1,400,000 (1,117,000)	1,077,000 (853,000)	45,800 (45,800)	1,270,000 (1,106,000)	953,000 (844,000)	40,900 (40,900)
VII	Noncombustion Organic Chemical Utilization	3,529,000	3,529,000	-	3,529,000 (3,162,000)	3,529,000 (3,162,000)	-	2,868,000 (2,666,000)	2,868,000 (2,666,000)	-
VIII	Agricultural & Forest Products	508,000	508,000	3,324,000	508,000 (371,000)	508,000 (371,000)	3,324,000 (582,000)	504,000 (367,000)	504,000 (367,000)	3,300,000 (576,000)
IX	Open Burning Sources (agricultural & prescribed forest burning)[d]	3,010,000	3,010,000	973,000	-	-	-	-	-	-
X	Natural Sources[e]	85,300,000	9,100,000	1,500,000	-	-	-	-	-	-
XI	Solid Waste Disposal[f]	2,690,000	2,443,000	640,000	115,500 (115,500)	115,500 (115,500)	108,800 (108,800)	64,800 (64,800)	64,800 (64,800)	75,000 (75,000)
XII	Municipal Sewage Disposal[g]	-	-	-	-	-	-	-	-	-
XIII	Other Sources[h]	917,000	917,000	234,000	-	-	-	-	-	-
		106,548,900	25,212,400	7,040,400	14,338,000 (4,882,500)	9,542,200 (4,618,500)	3,832,200 (736,600)	10,643,800 (4,319,800)	7,220,300 (4,057,800)	3,654,100 (691,900)
	Emissions from Processes Amenable to Controls / Emissions from All Processes, percentage				13.5	37.8	54.4			
	Emissions from Processes Amenable to Control with Afterburners / Emissions from All Processes, percentage				(4.6)	(18.3)	(10.5)			
	Controllable Emissions from Processes Amenable to Controls / Total Emissions from Processes Amenable to Controls, percentage							74.2	75.7	97.4
	Controllable Emissions from Processes Amenable to Afterburner Controls / Total Emissions from Processes Amenable to Afterburner Controls, percentage							(88.5)	(87.9)	(93.9)

---

a) Data compiled from "Hydrocarbon Pollutants from Stationary Sources", EPA Report No. EPA 600/7-77-110, September, 1977

b) Emissions are mostly methane and considered a low pollution hazard

c) Controllable fraction is IC engine sources

d) Only control is elimination of source

e) Not controllable

f) Except for incinerators, only control is elimination of sources (uncontrolled open burning)

g) No appreciable organic air emissions

h) Major sources are destructive fires and are not considered controllable

DEVELOPMENT OF EMISSION-CONTROL METHODS FOR LARGE-BORE STATIONARY ENGINES

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ABSTRACT

The research work presented herein was undertaken in order to develop combustion modifications which substantially reduce NO_x emissions of large-bore engines, and which result in equivalent or lower fuel consumption and carbonaceous emissions. The scope of the project covers NO_x control technology for diesel and spark ignition engines, bore sizes ranging from 8 to 20", and both 2 and 4 cycle charging methods.

In Phase I, a compendium of 40 emission control concepts was prepared, including methods which have shown promise for automotive engines. We also developed new methods using fundamental assumptions about the pollutant formation processes in spark ignition and diesel combustion environments. In Phase II, a ranking procedure was used to screen down the list to those concepts that are most promising and, therefore, suitable for testing in Phase III. The primary tool used in Phase II was a mathematical simulation of the combustion and pollutant formation process in spark gas engines. Also important in the selection was the practical feasibility, side effects, retrofit feasibility, and relative cost of each concept.

In Phase III, under major subcontracts, Cooper Energy Services and Fairbanks-Morse will use single and dual cylinder laboratory engines to test selected emission control methods. In Phase IV, the methods which prove effective in the laboratory will be applied to several engines operating in the field. Systematic measurements of BSFC and emissions will be made over an extended period in order to demonstrate the level of NO_x reduction and the reliability of the control technology. The program will conclude with an assessment of the costs and benefits of widespread adoption of the control methods.

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ACKNOWLEDGEMENTS

The project team at Arthur D. Little, Inc. includes Elia Demetri, W. David Lee, Philip G. Gott, Donald Hurter, William Raymond, and Arthur Fowle. Professor William J. McLean of Cornell University directed the development of the combustion model for spark ignition engines. Also we wish to acknowledge the assistance of Dr. Douglas Taylor of Ricardo, Professor Adel Sarofim of Massachusetts Institute of Technology, Professors Philip Myers and Gary Borman of the University of Wisconsin, Professor Lou Conta (University of Rhode Island), Charles Netwon and Eugene Kasel of Fairbanks-Morse, and Fred Schaub and Mel Helmich of Cooper Energy Services.

The Diesel Engine Manufacturers Association (DEMA) has been cognizant of major aspects of this project starting with the initial proposal in 1976. DEMA not only has an obvious interest in engine emissions R&D, but also is uniquely able to assist the program based on practical experience.

SECTION 1

INTRODUCTION AND ENGINE DESCRIPTION

INTRODUCTION

The effort described herein deals with a category of stationary combustion engines which is generally out of the public eye but which contributes a significant fraction of the total air pollution burden, particularly to the NO_x total. There are currently over 10,000 stationary reciprocating engines operating in the U.S. in the 8-20" bore class, ranging from 80 to 700 HP/cylinder. The estimated annual fuel use by these engines is 1.1 quads, or 1.5% of the U.S. energy budget; however, the cumulative NO_x emissions are disproportionately high at about 5% of the U.S. total. Among the many applications of these large reciprocating engines, the most significant single category is the gas pipeline compressor function, typically a two-cycle turbocharged engine of 14-20" bore. Other important categories in terms of fuel use are the gas gathering and electric generator engines of somewhat smaller bore size. Table I summarizes the population of stationary large-bore engines.

All of the combustion conditions which make large-bore engines relatively highly efficient prime movers (some operate as low as 6000 Btu/BHP-hr) also make them produce relatively high levels of NO_x (about 4 lb/MMBtu): high flame temperature due to compression preheating, low wall heat losses, air-fuel ratio in the range most conducive to NO formation, and relatively extended duration at peak temperature due to low rpm. Interest in developing emission controls for large bore engines has recently intensified, as evidenced by the proposed EPA standards (equivalent to about 9 gNO_x/BHP-hr, or a 30% reduction from current levels) and by recent hearings of the California Air Resources Board related to even lower target NO_x levels. The present study is part of a long-range research program to develop combustion modifications for large bore engines for the 1985-1990 time frame.

Large-Bore Engine Characteristics

The combustion process of the large-bore engines being considered here is distinguished from that in the more familiar spark ignited automotive engine by several characteristics. First, both the speed (300-1200 RPM) and the cylinder dimensions (8-20" bore) of the large-bore engines are a factor of 3 to 7 different than automotive engines. Second, most low-speed spark-ignited engines employ direct cylinder injection of gaseous fuel. Since gas injection terminates only 60° CA before TDC, the possibility of imperfect fuel/air mixing must be considered. The limited available information on the details of the combustion process indicates that the pre-combustion mixture of air, residual burnt gases, and gaseous fuel is not uniform, but rather exhibits a spatial non-uniformity in fuel-air ratio which we refer to as "unmixedness." Although the charge is non-uniform, all portions of the charge are reached by the flame, leaving little unburned fuel. Hydrocarbon emissions are typically only 0.5-2.0 g/BHP-hr for spark gas engines. In the two-stroke engines, the swirling flow of low turbulence level (induced by the loop scavenging process) apparently assures that the mixing process is adequate if not complete.

Most two-cycle gas engines are adjusted to operate quite fuel lean by automotive engine standards; a mean fuel-air equivalence ratio between 0.7 and 0.8 is not uncommon. Reasons for using lean mixtures are improved cycle efficiency (lower SFC), moderation of pressure rise rates, and prevention of knock due to end gas autoignition. Such lean mixtures, of course, require a strong ignition source, particularly for methane fuel. In this regard it is interesting to speculate that the presence of richer than average regions of charge near the spark source due to unmixedness may promote more reliable ignition.

While the low turbulence intensity and large cylinder size in the gas engines lead to combustion durations in the range 5-15 msec, the relative duration in crankangle degrees is only 10-30° CA because of the low engine speed. Thus, at 330 RPM, a 20° CA combustion duration and a spark advance of about 10° BTDC is typical. The advance in spark timing is limited by the onset of knock, which is attributed to small quantities of less knock-resistant higher paraffins (such as butane) in natural gas, which is largely composed of highly knock resistant methane.

With respect to formation of nitric oxide, the conditions described above could hardly be better selected for maximum NO production. Experience with automotive engines indicates that maximum NO is to be expected for the lean high-temperature mixture conditions produced in the large-bore engines. Futhermore, the low surface-to-volume ratio results in near-adiabatic conditions. The low engine speed allows sufficient time at high temperatures for the rate controlling $O + N_2 \rightarrow NO + N$ reaction of the Zeldovich mechanism to produce nitric oxide concentrations which approach equilibrium levels in the hottest portions of the charge. As in the automotive engine, these high NO_x concentrations are preserved during the expansion stroke because the NO decomposition reactions are relatively slow in the temperature range which is characteristic of expansion.

The current NO_x emissions of large-bore engines are clustered in the 12-15 g/BHP-hr range. This is equivalent to approximately 4 lb/MM Btu, which is unusually high for combustion devices, as illustrated in the following table:

NO_x EMISSION RATE OF LARGE BORE ENGINES
COMPARED TO OTHER COMBUSTION DEVICES

Device	Relative Fuel Use, Nationwide	Typical NO_x Emission (lb/MM Btu)
Large Bore Engines	1	4
Automotive SI Engine	14	2
Coal Fired Utility Boilers	15	0.7
Industrial & Commercial Boilers (Oil-Fired)	17	0.4
Industrial Furnaces (Gas-Fired)	12	0.3
Residential Furnace and Water Heater	10	0.1

The factors which couple high NO_x to high efficiency comprise the basic constraint on emission control techniques: efforts to reduce NO_x are likely to reduce efficiency as well, unless efforts are made to keep heat release near TDC. Simple methods to reduce NO_x, such as EGR and spark retard, are known to have BSFC penalties; the effort of the proposed program will be on developing NO_x control techniques which potentially do not have a BSFC penalty, such as modified fuel/air preparation, chamber shape modifications, and NO_x decomposition.

Program Methodology

The work described herein was undertaken in order to develop and screen potential emission control concepts prior to engine tests. The aim of Phase I has been to generate an investory of existing and new emission control concepts for large-bore stationary engines. The scope included both spark gas engines as well as diesel engines in the medium speed (300--1200 rated RPM) range. This inventory of concepts was then subjected to a critical screening process in Phase II, where the concepts were compared based on their potential merit, considering emissions reduction, effect on brake specific fuel consumption (BSFC), practical feasibility, and ultimate cost to users. In order to estimate the effect of various NO_x control measures on both emissions and fuel consumption, it was necessary to construct a model which would adequately account for the conditions peculiar to large-bore spark gas engines. It was considered particularly important to include the effect of unmixedness or dispersion in the local fuel-air ratio, as this feature of large-bore engines had not been generally included in models of automotive engine combustion. A description of the model is presented in Wilson et al(1979).

Since the focus of the program is on exploring the feasibility of achieving substantial NO_x reductions for future engine designs, emphasis was placed on combustion modification concepts, such as torch ignition, stratified charge, water-fuel emulsions and novel injection systems rather than on "external" adjustments alone, such as EGR and timing. Exhaust gas treatment and combinations of concepts were also studied. In Wilson (1978), each concept for emission control is described with schematics showing the manufacturers' preferred and alternative configurations.

SECTION 2

EMISSION CONTROL MECHANISMS FOR SPARK IGNITION ENGINES

COMBUSTION CONDITIONS AND NO_x PRODUCTION IN SPARK-IGNITION (SI) ENGINES

In a large-bore SI engine, the main portion of the charge can be assumed to be premixed, and therefore combustion is believed to occur by propagation of a turbulent flame front outward from the spark. The burned gas at the flame front is thought to engulf eddies of fresh mixture by a turbulent mixing process. Ignition sites appear on the boundaries of the fresh mixture eddies and then a flame can be considered as traveling across each eddy at the laminar flame speed, as suggested in Figure 2-1. The heat released by this process serves to propagate the flame in two ways: (a) the turbulent mixing of reactants into products is enhanced by sudden expansion of eddies following ignition and (b) the heated combustion products ignite entrained elements of reactants.

The NO_x formed in an SI engine is essentially related to the conditions in the hot gas left in the wake of the flame front. Each successive element of combustion gas which is produced by the passing flame front generates NO_x depending on its initial flame temperature, fuel-air ratio, rate of compression heating, and rate of quenching. Figure 2-2 shows the initial NO production rate in ppm/msec as a function of temperature and fuel-air ratio. NO_x formation rates of 100 ppm/msec or greater are practically unavoidable (for at least very short times) because, as shown in Figure 2-2, the adiabatic flame temperatures predicted for typical engine conditions (800°K compression temperature, 40 atm) are on the order of 2500°K depending on fuel-air ratio. The nitric oxide accumulated by a given element of combustion products will be the time-integral of the formation rate along a temperature history, as depicted by the path shown in Figure 2-2. Most critical to the cumulative NO emission are two factors:

- The conditions (flame temperature, fuel-air ratio) at which each mixture element ignites.
- The subsequent temperature history, particularly the compression heating (which is the greatest for earliest burned elements), and the subsequent rate of cooling.

The first few elements to burn are thought to produce relatively high NO_x, because they subsequently undergo greatest compression heating and experience a relatively long residence time at high temperature.

Under certain conditions, some decomposition of NO can occur during the expansion stroke. This decomposition is driven by the difference between actual NO level and equilibrium NO level.

GENERAL APPROACHES TO NITRIC OXIDE CONTROL

The various techniques for reducing the level of NO_x emitted by an engine can be visualized in terms of the (ϕ, T) histories of burned products. In Figure 2-3, three basic categories of NO_x-suppression techniques are illustrated: stratified combustion, lean combustion, and temperature suppression.

Stratified Combustion

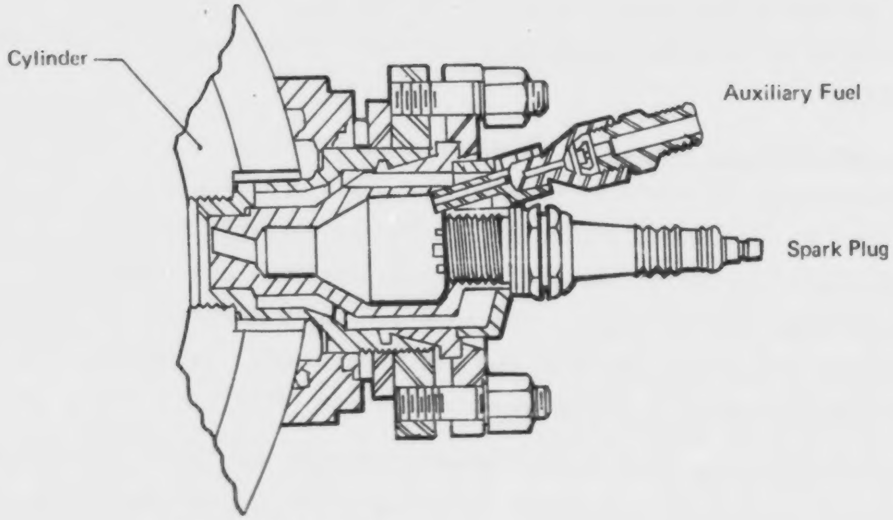
In the stratified charge engine, the fuel is intentionally maldistributed or layered so that one segment of the mixture is lean ($\phi \approx 0.6$) and the other segment is rich ($\phi = 1.2$-1.3) during and just after the combustion process. Both segments are outside of the fuel-air equivalence ratio range $\phi = 0.8$ to 1.1 where NO_x production rates are 100 ppm NO_x/msec or greater. After combustion the burned gas segments intermix at a point well into the expansion stroke, producing the overall average value of ϕ. It is also important to burn the $\phi > 1.2$ segment first. The fuel-rich segment is set-up in the vicinity of the spark plug, so that the elements burned as the flame moves out from the spark are subject to low-NO_x formation rates (due to lower temperature and the lack of oxygen). Igniting this fuel-rich segment has a second advantage in that righ gas is more suitable for ignition than the overall charge if it were homogenous.

For the large bore engine, stratified charge is not expected to produce dramatic NO_x reductions because in current turbocharged engines the mixture is typically already outside of the critical $\phi = 0.8$-1.1 range. For example, starting from overall $\bar{\phi} = 0.75$, division into segments of 1.0 and 0.5 actually is expected to increase the NO_x emission. This is borne out by the predictions shown in Figure 2-4, which show a 27% NO_x increase for a 1.0-0.5 stratified mixture. The explanation is that the $\phi = 1.0$ segment adds more NO_x than is subtracted by the $\phi = 0.5$ segment. Blumberg (1972) corroborates these predictions with his model for small-bore engines, showing an increase of NO_x from 1500 ppm to as high as 2600 ppm for the 0% EGR case. Blumberg emphasizes that for lean-burn engines (such as the large bore spark gas type), stratification in fact becomes quite risky since NO_x can increase from improper stratifications which place near-stoichiometric segments in the first elements to burn.

Lean Combustion

Here the mixture is made to burn more fuel lean than normal ($\phi = 0.6$-0.7), which reduces the exposure of products to high NO-production rates accordingly. Available data suggests that a 40% NO_x reduction can be obtained by shifting the equivalence ratio from 0.75 to 0.65. The computer model developed to simulate combustion and NO_x formation in a large-bore engine was applied to a representative 20" bore engine in order to project the NO_x reductions achievable by increasing air-fuel ratio. The results are shown in Figure 2-5. As the air is increased from a normal condition at $\phi = .78$ to the "lean" condition at $\phi = .70$, the NO_x was predicted to drop from 3600 to 2600 ppm (28%). Experimental data from a single cylinder engine generally agrees with the predictions on the effectof excess air, but show a slightly greater reduction (35%).

The basic problem does not lie with the technology for making the mixture lean. An increase in the trapped air mass in the cylinder can be accomplished by increasing the degree of turbocharging. The air mass is increased rather than the fuel decreased in order to maintain BMEP or engine power. The basic problems with burning mixtures at $\phi = 0.6$ to 0.7 are stimulating reliable ignition (preventing misfire) and limiting the combustion duration. Therefore, each lean combustion concept involves a remedial measure for misfire and combustion duration. One of the most promising concepts for minimizing combustion duration under lean conditions is torch ignition, as illustrated in the following figure:



Source: Fairbanks-Morse.

FIGURE 3–4 TORCH IGNITION DEVICE FOR 2 CYCLE GAS ENGINE

Temperature Suppression

By cooling the intake air or adding inerts, such as water vapor or EGR, the peak flame temperature is reduced. This shifts the NO-production rates to lower values. These concepts are also applicable to diesel engines.

NO_x Decomposition

In addition to these three categories of techniques, NO_x decomposition can be considered. In the late stages of expansion and in the exhaust manifold, the equilibrium NO-levels are less than a few hundred ppm. The actual NO level (which is a few thousand ppm) is relaxing toward this level at a negligible rate. Methods for enhancing or catalyzing the NO decomposition process were considered. There are two distinct approaches to carrying out NO_x decomposition, each of which has been demonstrated for industrial boilers in Japan (where NO_x regulations are severe):

- Gas Phase Reaction: Conducted at high-temperature (1000-(1400°K or 1300-2000°F). This approach if applied to engines would require no catalyst, but would require costly reheating of the exhaust gas to at least 1300°F, particularly for 2-stroke engines. Expected NO_x reductions are in the 50-60% range for a 2:1 NH_3/NO_x mole ratio, according to Lyon and Longwell (1976) and Muzio et al (1977).

- Catalyst-Induced Reaction: Conducted at moderate temperature (500-650°K or 440-700°F). This approach if applied to engines would require costly catalyst replacement every 1-3 years (about $10/HP for platinum) and would entail some pressure drop. The expected NO_x reductions, however, are higher (80-90%) and 25% less ammonia is needed (NH_x: NO_x ratio of 1.5), according to Bartz (1977).

Both approaches require a costly ammonia storage and injection system. At this writing, these approaches are receiving attention in connection with possible California NO_x standards.

The following table summarizes the emission control methods which have been compiled for large-bore spark ignition engines in this study.

Category	Concepts Considered Potentially Practical	Concepts Considered Impractical
Lean Combustion	Torch ignition Multiple spark plugs Increased turbulence High-energy spark Diesel fuel ignition Feedback control w/O_2 sensor	Shock wave ignition Hydrogen enrichment Catalytic piston Optical ignition Pyrophoric jet ignition
Stratified Combustion	Divided chamber stratification Open chamber stratification (Auxilliary injection port or carburetor) Degraded mixing	
Temperature Suppression	Charge refrigeration Retarded timing EGR	LNG injection Intake water injection Increased engine speed
NO_x Decomposition	Ammonia reduction agent Nitrogen plasma injection Ozone injection Chemical absorption	Metal exhaust catalyst Catalytic piston Modified cooling rate

IMPACT OF COMBUSTION MODIFICATION ON FUEL CONSUMPTION

Stratified combustion, lean combustion, and temperature suppression (the basic techniques of NO-suppression illustrated in Figure 2-3) all have the effect of (a) delaying the establishment of a vigorous propagating flame and (b) lowering the rate of combustion which, in turn, can adversely affect BSFC and carbonaceous emissions. The timing of an SI engine is normally adjusted so that the peak pressure occurs just after top center, so that the heat released by the burned fuel appears as close as possible to top center (before piston movement). That portion of the heat which is released to the gases during the expansion stroke (denoted "late burning") does not contribute fully to the work. The heated gases from "late burning" expand through only a portion of the volume change. The efficiency loss due to extended combustion duration has been studied by Lyn (1960). Figure 2-6, taken from Lyn's findings, suggests that the efficiency penalty is about two percentage points for each 10°CA extension of the combustion duration. Partial compensation for this penalty can be obtained by advancing the ignition point.

The significance of BSFC degradation cannot be overestimated, since the fuel costs for large-bore engines represent a substantial portion of the overall costs. In this project we have arbitrarily adopted a goal of achieving maximum NO_x reductions within a 3-4% BSFC penalty. The basis for this goal is the engine adjustments such as retarded timing alone can substantially reduce NO_x but only with a 6-8% fuel penalty [see Youngblood et al. (1978)].

SECTION 3

EMISSION CONTROL MECHANISMS FOR DIESEL ENGINES

COMBUSTION CONDITIONS AND NO_x PRODUCTION IN DIESEL ENGINES

The formation of NO_x in a diesel engine must be described in terms of the sequence of combustion processes which occur as the spray mixes and burns in the chamber. Following the excellent descriptions put forth by Lyn (1962) and Austen and Lyn (1960), the four stages of combustion are as follows:

- Compression and preignition mixing
- Ignition and flame propagation
- Spray combustion
- Residual combustion and mixing

These processes are more complex than in the premixed charge of an SI engine, because of spray formation, fuel-air mixing and radiation. In the following sections, we will describe each stage and will comment on the factors affecting NO_x-formation.

Compression and Preignition Mixing

At the start of fuel injection, the air has been heated to 800°K* by compression and endowed with turbulence due to piston motion, "squish," and swirl. Liquid fuel is injected into the large-bore diesel starting at about 10°BTDC*, and ignition occurs about 7°CA (3 msec)* later, after a portion of the fuel has evaporated, mixed with air, and undergone a chemical ignition delay which is more or less characteristic of the compression temperature. According to Andree and Pachernegg (1969), ignition occurs whenever the evaporated fuel has been exposed to a temperature surplus (over the reference ignition temperature, for a certain integral time ($\int \Delta T \, d\tau \geqq 335$°K-msec).

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*Temperatures and crank angles quoted are typical; actual values for specific engines vary widely.

There is no NO_x formation up to the ignition point; however, the subsequent NO_x production is highly influenced by such factors as compression ratio, charge precooling, timing of injection, and turbulence.

Ignition and Flame Propagation

The first phase of combustion is relatively brief, having a high heat release rate which is often termed the "spike" on a plot of heat release rate versus time (see Figure 3.1). The reason that this first phase proceeds so swiftly is that multiple ignition sites occur and flame kernels propagate through all the portions of the chamber which contain a premixed flammable air fuel charge. The high-speed photographs of Rife and Heywood (1974) suggest that first combustion occurs in the zone where the vapor plume impinges with the wall. For the large-bore diesel we estimate that about 5-10% of the heat release is "prepared to burn" (evaporated and mixed) and attributable to the "spike." A disproportionate amount of NO_x, however, is thought to be formed in regions ignited by the spike for the same reasons that the zones near the spark plug of SI engines exhibit high NO_x:

- Earliest-produced combustion products have longest residence time at high temperature.
- Subsequent compression heating of earliest zones leads to elevated temperature and NO_x rates.
- The nature of premixed combustion makes the spike-affected zones more adiabatic; i.e., the quenching rate is limited because neighboring elements are burnt products rather than cooler air.

If a large number of fuel elements are all suddenly ignited and burned, the resulting high rate of pressure rise increases NO_x by compression heating of burned gases.

The first guideline for avoiding high NO_x combustion conditions, therefore, is to minimize the heat release by the spike; that is, to minimize the accumulated premixed fuel vapor which is available for burning at the time of first ignition. Retarded fuel injection accomplishes this, because the higher compression temperatures at the retarded crank angle shorten the ignition delay. Fumigation, water emulsion, pilot injection, and cetane number modifications to the fuel are optional methods which can be considered to shorten ignition delay. Changes in fuel injectors will also affect the amount of fuel vapor available at the fuel-air ratio most conducive to ignition ($\phi = 1.1$).

Spray Combustion

Overlapping with the flame-propagation stage is the second stage of combustion which is characterized by mixing-controlled combustion of the fuel droplet array and unmixed fuel vapor pockets. These fuel elements are dispersed into the product/air mixture by swirl, by wall impingement, and by spray penetration; and they burn as fast as the droplet array can evaporate and find air. For the large-bore medium speed diesel engine, the photographic evidence of Taylor and Walsham (1970) suggests that the spray is vaporized completely before it reaches the wall, but that wall impingement of the vapor plume occurs. Rife and Heywood (1974) concluded that mixing (and therefore the combustion rate) is controlled by the rate of air entrainment by the fuel spray or vapor plume, rather than the ballistic dynamics of single droplets. The burning history of each fuel element resembles a downward ramp as shown in Figure 3-1. The cumulative heat released from all elements follows the familiar "Wiebe" rate, again shown in Figure 3-1. This stage of combustion lasts 10-15 msec, depending on the duration of fuel injection and the rpm.

Nitric oxide production during this mode of combustion depends on the distribution of fuel-air ratio within the various mixing-limited subzones, as illustrated in Figure 3-2. The mixing-limited subzones can be described by a fuel-air ratio distribution which presumably centers around stoichiometric fuel-air ratio, based on fundamental considerations about diffusion flames. The guideline for avoiding high-NO_x producing combustion conditions for this second stage of burning is to reduce the residence time of diffusion burned zones at temperatures above 2200°K. This can be approached through (a) reducing the post-combustion compression by retarded injection timing (coupled with increased mixing to insure that burning is complete prior to expansion), (b) forcing off-stoichiometric conditions such as occur in a prechamber engine, and (c) increasing swirl or turbulence to lean out the hot burnt gases with air.

Product-Air Mixing and Residual Combustion

In this final stage of combustion, the elements of combustion products continue to intermix with each other and with the remaining air in the chamber. Any residual fuel vapor, CO, or hydrogen found in elements with insufficient air are burned to CO_2 and H_2O in the process of mixing. The entire cylinder contents gradually become more uniform in excess air and temperature. A given burned gas element is quenched from 2600°K to below 2200°K in about 0.3 msec for the high-rpm engines simulated by Murayama et al (1977). During this 0.3 msec the NO_x is being produced at a rate of about 1000-3000 ppm/msec according to Figure 3-2.

NO_x production in this stage can be minimized by increasing the mixing rate so that relatively cool air can quench NO-production elements. It is important that this increased mixing rate be timed to occur after combustion rather than before combustion. Otherwise the heat release rate (and NO_x) in stage I will increase.

Summary

In summary, the unique combustion factors of large-bore diesel engines are as follows:

- Larger fuel orifices (.028" to .008") lead to larger drop sizes (SMD 22μ versus 15μ).
- More fuel jets (10 versus 4).
- Low Swirl.
- Less heat release in the premixed stage (spike)--approximately 10%.
- Longer duration at near-constant volume (600 rpm versus 2400 rpm).
- Lower compression ratio (12 to 14).
- Higher degree of turbocharging (up to 280 BMEP).
- Leaner full load operation ($\phi = 0.5$ versus $\phi = 0.7$).
- Peak pressure more critical (structural-welded, not cast).
- Lower surface area per unit volume--less specific heat transfer and larger quench fraction.

APPROACHES TO NO_x CONTROL FOR DIESEL ENGINES

To summarize the discussion up to this point, NO_x reduction is best achieved for both premixed (stage 1) and spray combustion (stage 2) by retarding the injection and compensating for this by reducing the ignition delay and increasing the mixing rate. Particular attention must be given to minimizing the time spent by the initial early-mixed charge at 2200-2600°K (near-stoichiometric flame temperature).

Approaches to NO_x control for diesel engines are in the table below.

Category	Concepts Considered Potentially Practical	Concepts Considered Impractical
Modified Fuel Injection	Water-fuel emulsion High injection rate Pilot injection	Bi-fuel injection Air-assist atomization
Altered Mixing Pattern	Prechamber-variable area Modified air motion/chamber shape Circumferential injection	Initial charge modification
Temperature Suppression	Charge refrigeration EGR/retarded timing	Intake water injection Increased engine speed
NO_x Decomposition	Ammonia reduction agent Nitrogen plasma injection Ozone injection Chemical Absorption	Metal exhaust catalyst Catalytic piston Modified cooling rate

Water-Fuel Emulsions

Data for small-bore diesel suggests that a 50% water/fuel ratio by volume can reduce NO_x in the exhaust by about 50% when injected as an emulsion (see Figure 3.3). Large-bore engines have not been operated at water-fuel ratios above 25%, however, the results are promising (see Figure 3-4). The introduction of water as an emulsion in diesel fuel is thought to lower NO_x production by the following mechanisms: First, latent heat and enthalpy is absorbed by the water, which lowers the local temperature of the products of combustion, lowering the rate of NO_x production. In addition, it has been hypothesized that the atomization process is altered by so-called "micro-explosions" of emulsified droplets. It is not clear to what extent this affects NO_x significantly.

Practical problems of emulsions which must be addressed include the following:

(a) The injection nozzle and cam must be changed if necessary to accommodate the increased volume of fluid to be injected, while maintaining injection duration and atomization quality. An increase in the number of holes is one option.

(b) Preventing "slugs" of water from separating out of the emulsion--Non-emulsified water in fuel systems has been responsible for severe damage to piston crown and cylinder heads.

(c) The effect of the fuel oil-water mixture on the durability of the engine and the availability of water is of concern. Lestz et al (1975) indicate little adverse effect in fuel components even at existing high water injection rates (300-500% water/fuel ratio). Demineralized water is required to minimize deposit build-up, apparently.

(d) Prevention of excessive cylinder pressure by retarded timing.

SECTION 4

COMPARISON OF EMISSION CONTROL METHODS

COMPARISON OF METHODS BASED ON POTENTIAL NO_x REDUCTION

Since the primary objective of the project is emissions reduction, the most critical consideration in comparing the candidate concepts is what fraction of the normal NO_x emission can probably be eliminated by each concept. Fuel consumption (BSFC), however, is also an important consideration. It is well established that 40% NO_x reduction can be achieved on most engines by applying methods which entail 6-8% BSFC penalty (e.g., timing and EGR). Therefore, in order to take BSFC into account while placing primary emphasis on NO_x, we have projected the maximum feasible NO_x reduction without exceeding 4% BSFC penalty. Based on underlying mechanisms of hydrocarbon and CO emissions, it is felt that 4% BSFC limit would also constrain these emissions to acceptable levels.

Three sources of information were used to project NO_x reductions:

- Emission test data for previous experimental attempts to apply the concepts to large and small-bore engines (50% weighting factor).
- Engineering judgment by those experienced in combustion and emissions control (25% weighting factor).
- Mathematical model predictions using a computer simulation of a large-bore spark gas engine (25% weighting factor).

These sources of information were weighted as noted and a consensus projection was reached in this manner. Table II lists the projected NO_x reductions using this procedure. Note that when engine data was not available, the model projections and "engineering judgment" received equal weight.

Excluding the four exhaust treatment methods, relatively little variation apparently exists in the NO_x-reduction potential, based on Table II (the range is 16 to 37%). There were no concepts in the range of 40-60% reduction. Reaching this target, however, is still a rational goal for combustion modification, because:

- These projections are reasonably conservative. For example, the lean operation could have been taken at $\phi = 0.60-0.65$ where greater NO_x reductions would be expected. Small-bore engine applications and isolated large-bore experiments have shown 50-70% reductions are possible.
- Combinations of techniques may prove feasible.

COMPARISON OF EMISSION CONTROL CONCEPTS BASED ON FEASIBILITY AND COST

In order to identify those concepts with enough promise to be advanced to engine experimentation, in addition to NO_x reduction projections, a critical review was made of each concept which involved quantitative cost factors, such as:

- added costs of the modified engine;
- cost of developing the concept;
- added maintenance costs;
- other added costs of operating the modified engine.

In addition, there are qualitative factors having to do with practical feasibility which were also considered, as follows:

- applicability of the concept to important engine classes and major manufacturers' engines (weighting factor, WF = 7);
- practicality of retrofitting engine in the field (WF = 5);
- adverse side effects anticipated, such as noise, corrosion, odor, or poor starting (WF = 3);
- the need for special materials (WF = 2);
- operating complexity, reliability, and the feasibility of unattended operation without unscheduled breakdown (WF = 7);

- effect of reciprocating engines' salability in competing against gas turbines (WF = 3);
- whether manufacturers are already implementing the concept on their own on a proprietary basis, so that EPA efforts would be redundant (WF = 5).

A ranking of concepts according to feasibility was carried out by a formal procedure. Seven engine manufacturers and Ricardo were contacted for practical engineering judgments regarding the feasibility of 22 emission control concepts. Weighting factors were applied to each criterion and the scores were normalized so as to permit an overall figure of merit or "feasibility index." Strengths and weaknesses of each concept were identified relative to other concepts. The principle findings were, as follows:

- There was no concept which could absolutely not be implemented on at least one engine type.
- There was relatively little variation in overall feasibility among the 22 concepts. Numerically, on a scale of 0 to 1.00, the feasibility index ranged from .62 to .81 for spark ignition engines and from .56 to .89 for diesel engines. Each concept appears to have offsetting strengths and weaknesses.
- Engine manufacturers' greatest concerns centered on unscheduled downtime (reliability), requirements for maintenance of sophisticated electronic components, and avoiding storage and handling of additional fluids (e.g., liquid nitrogen, demineralized water for emulsion, or chemicals for scrubbing).

In summary, apart from cost-effectiveness considerations, provided that concerns about reliability and "second fluid" can be met to manufacturers' satisfaction, it would be difficult to rule out any of the concepts based on feasibility or lack thereof.

OVERALL RANKING CONSIDERING NO_x REDUCTION POTENTIAL, ADDED COSTS, AND FEASIBILITY

The pertinent figures of merit for each of the concepts are presented in Table III by engine type. Note that there are certain concepts with noticeable discrepancies when ranked by different criteria, for example:

	NO_x	Cost	Feasibility
• Torch Ignition	1	6	5
• Turbulence	7	1	1
• Degraded Premix	5	2	7
• Charge Refrigeration	2	10	1
• Emulsion	1	6	5

This points out the need to balance or tradeoff the three considerations in deciding which concepts are to be tested in Phase III. It must be emphasized that the feasibility indices are quite closely grouped, and both the NO_x and the cost projections are uncertain to about a factor of a third the value quoted in either direction (i.e., 30% means 20-40%). An additional consideration is the relative cost to experiment with the concept in Phase III.

Figures 4-1, 4-2 and Table IV combines the figures of merit in three ways, each of which may be useful to the decision:

- Cost-effectiveness ($\Delta NO_x \div \Delta cost$)--How much NO_x is reduced for the added operating cost? Figure IV displays the results.
- Feasibility-effectiveness (ΔNO_x x feasibility)--How much is NO_x reduced for a given degree of practical feasibility? Figure 4-2 displays this index.
- All factors (ΔNO_x x feasibility $\div$ Δcost)--This is one of many conceivable ways to combine the three indices and can be recommended only for its simplicity. Table IV displays this index.

The bias of the third (combined) figure of merit can be assessed, by examining the rankings of the five concepts mentioned above, as follows:

	NO_x Alone	Cost Alone	Feasibility Alone	All Factors Combined
• Torch ignition	1	6	5	2
• Turbulence	7	1	1	1
• Degraded premix	5	2	7	4
• Charge refrigeration	2	10	1	8
• Emulsion	1	6	5	5

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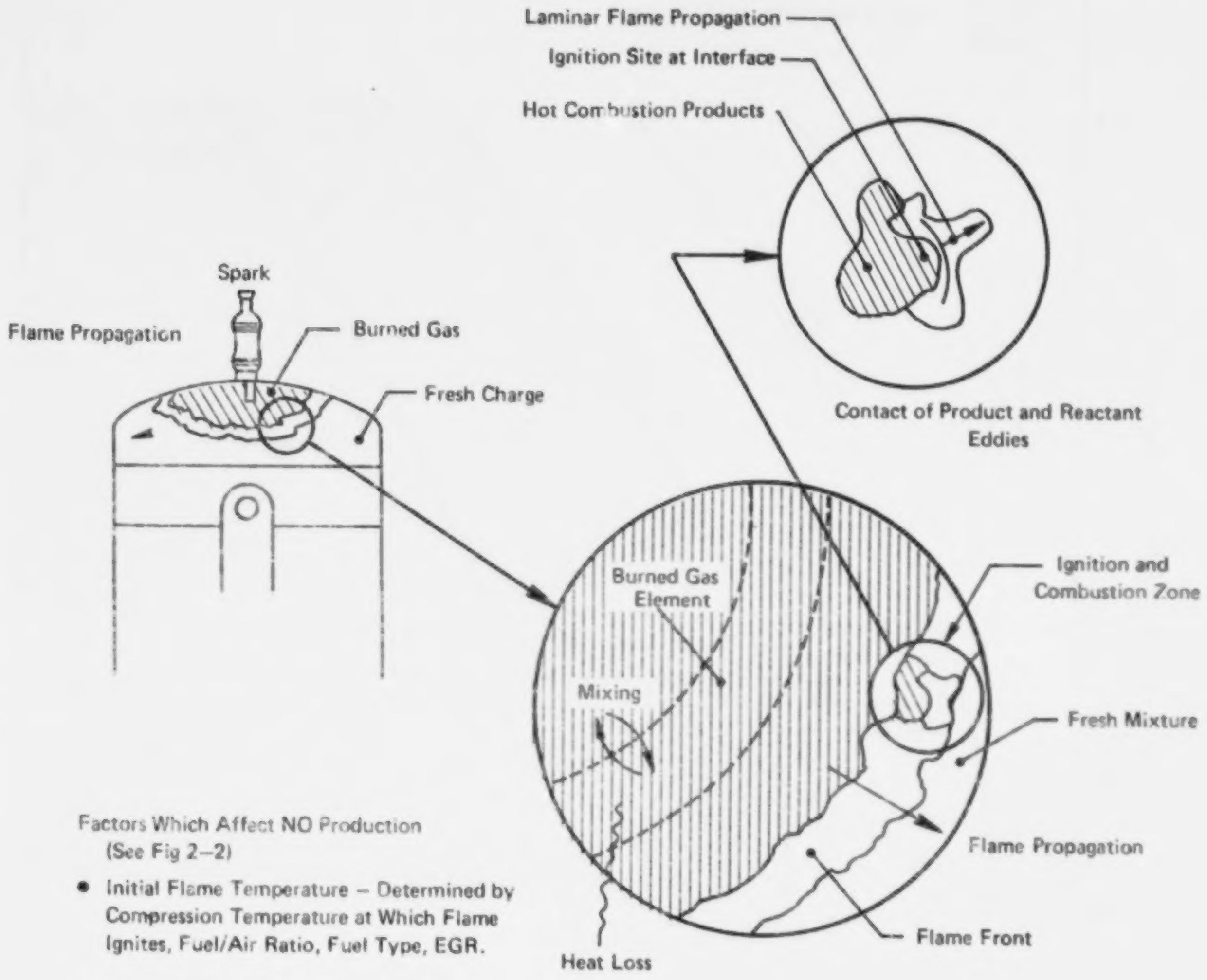

Laminar Flame Propagation
Ignition Site at Interface
Hot Combustion Products
Contact of Product and Reactant Eddies
Spark
Flame Propagation
Burned Gas
Fresh Charge
Burned Gas Element
Mixing
Ignition and Combustion Zone
Fresh Mixture
Flame Propagation
Flame Front
Heat Loss

Factors Which Affect NO Production
(See Fig 2–2)

- Initial Flame Temperature – Determined by Compression Temperature at Which Flame Ignites, Fuel/Air Ratio, Fuel Type, EGR.
- Temperature History – Determined by Compression Heating Due to Combustion, Wass Heat Loss, Mixing With Cooler Gas, and Cooling Due to Piston Motion.

Figure 2–1. Combustion and No_x formation in SI engines.

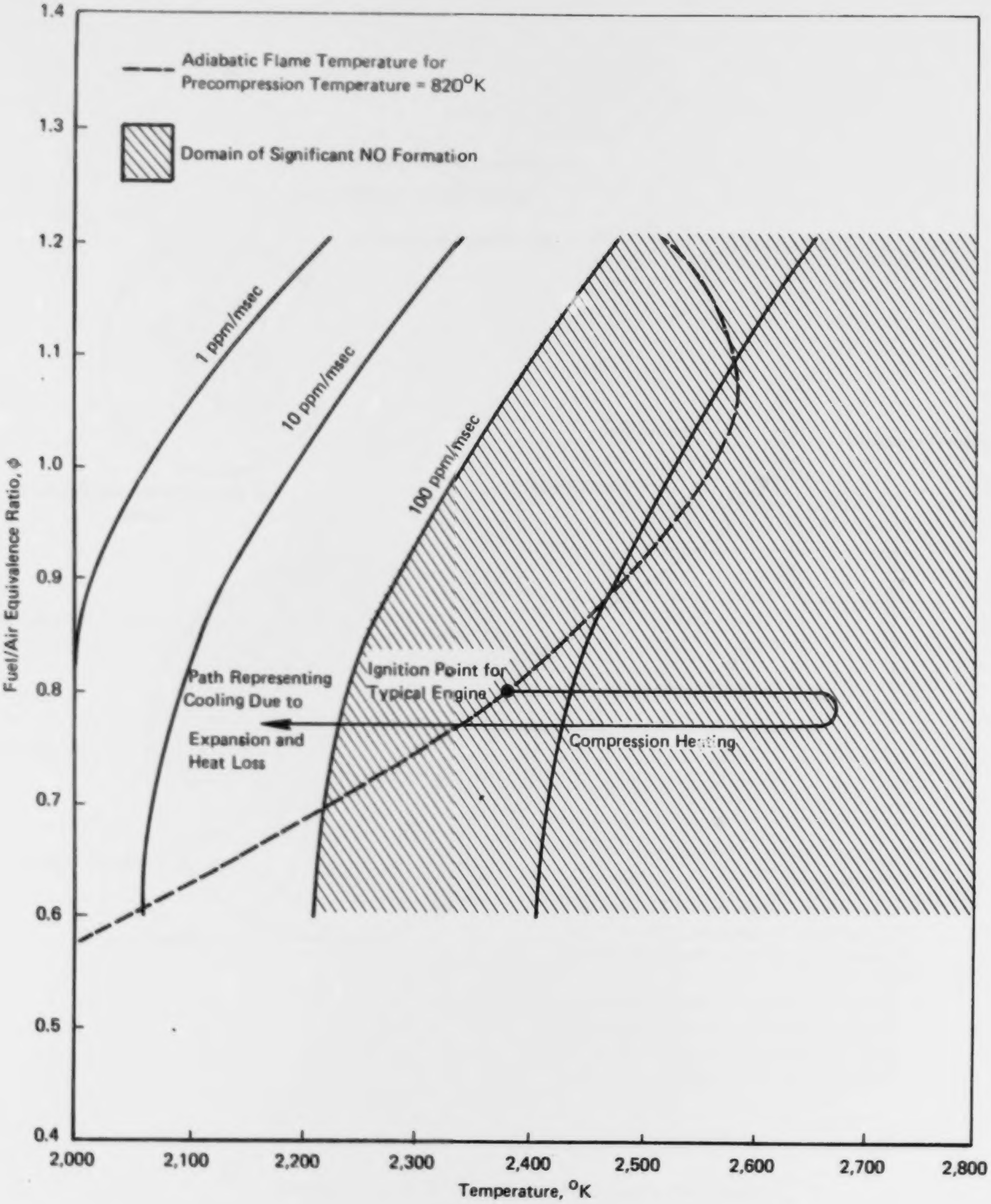


Figure 2–2. Initial nitric oxide formation rate . (40 atm pressure)

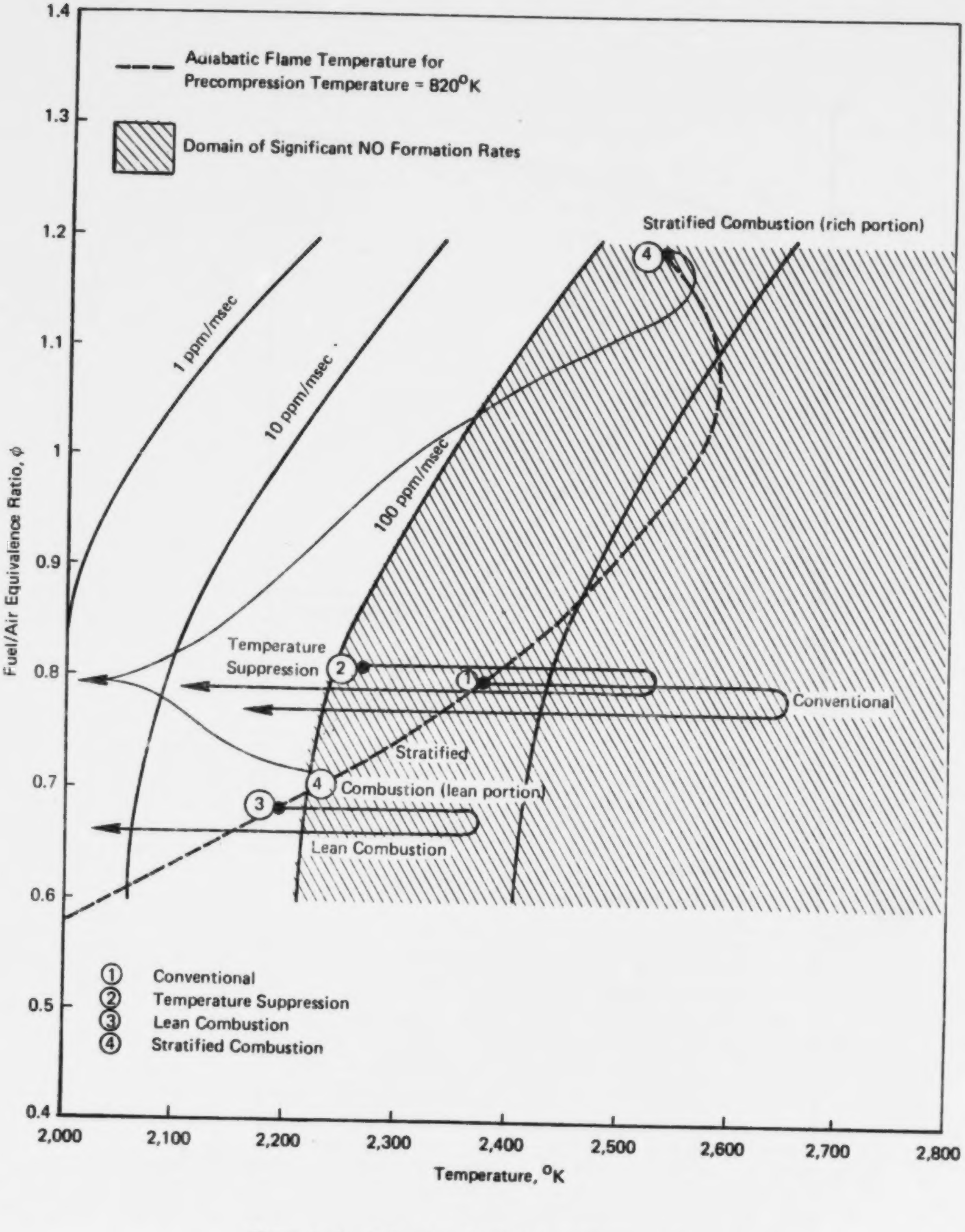


Figure 2–3. Basic categories of nitric-oxide supression.

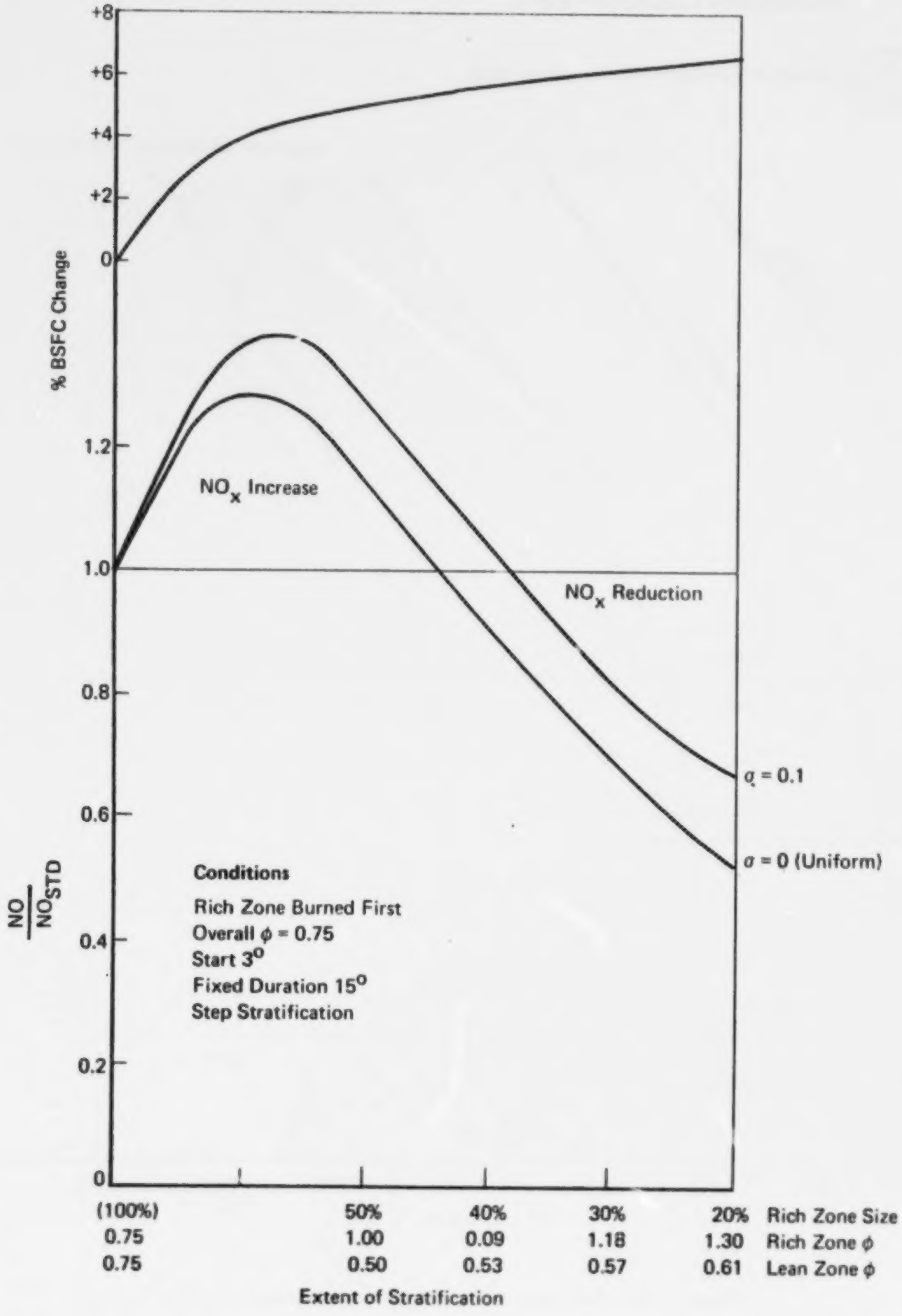


FIGURE 2-4: **PREDICTED EFFECT OF STRATIFIED CHARGE**

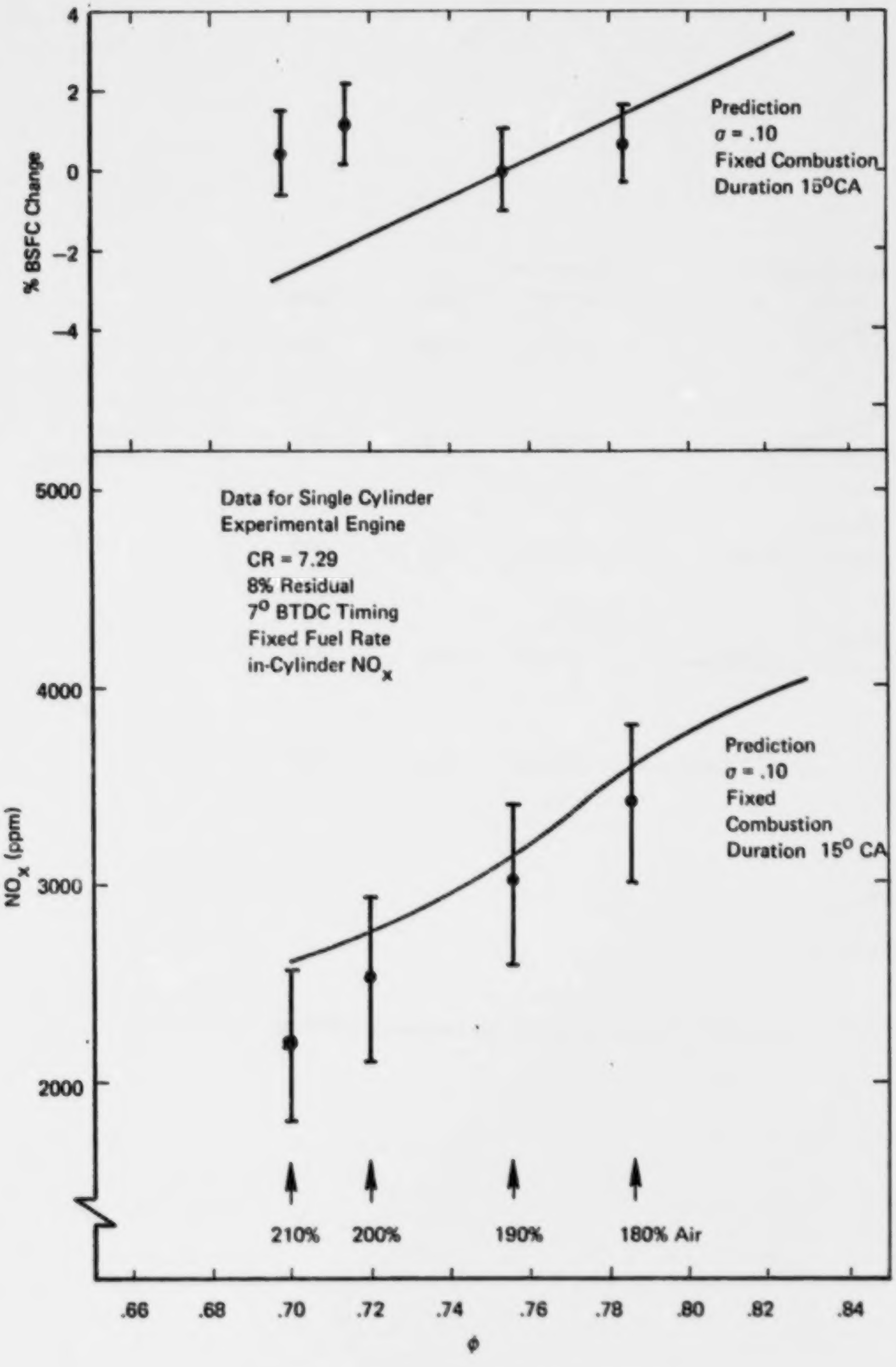
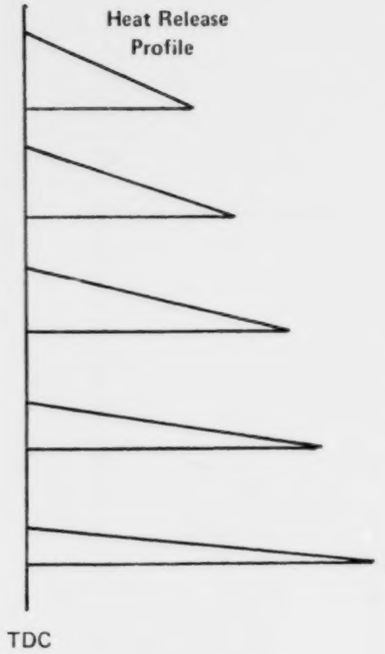


FIGURE 2-5: **EFFECT OF EXCESS AIR**
(TWO STROKE SPARK GAS ENGINE)


Heat Release Profile
TDC

Combustion Duration	η_{ind}	η_{brake}*	
30°C.A.	54.9%	44.9%	Range of Typical Engines
40°C.A.	52.5	42.5	
50°C.A.	50.8	40.8	
60°C.A.	48.6	38.6	
70°C.A.	46.4	36.4	

*Assuming $\eta_{brake} = \eta_{ind} - 10\%$

Source: Lyn (1960).

Figure 2–6 **Effect of combustion duration on efficiency.**

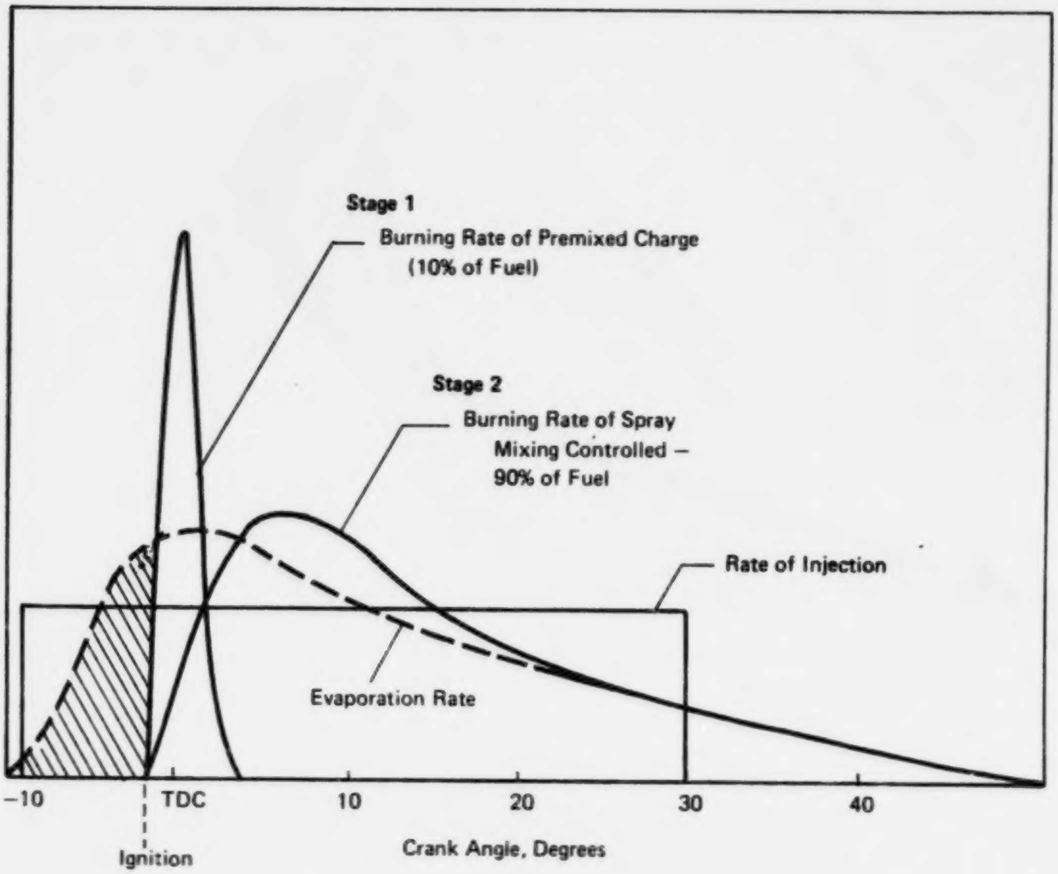


Figure 3–1. Illustration of two stages of combustion.

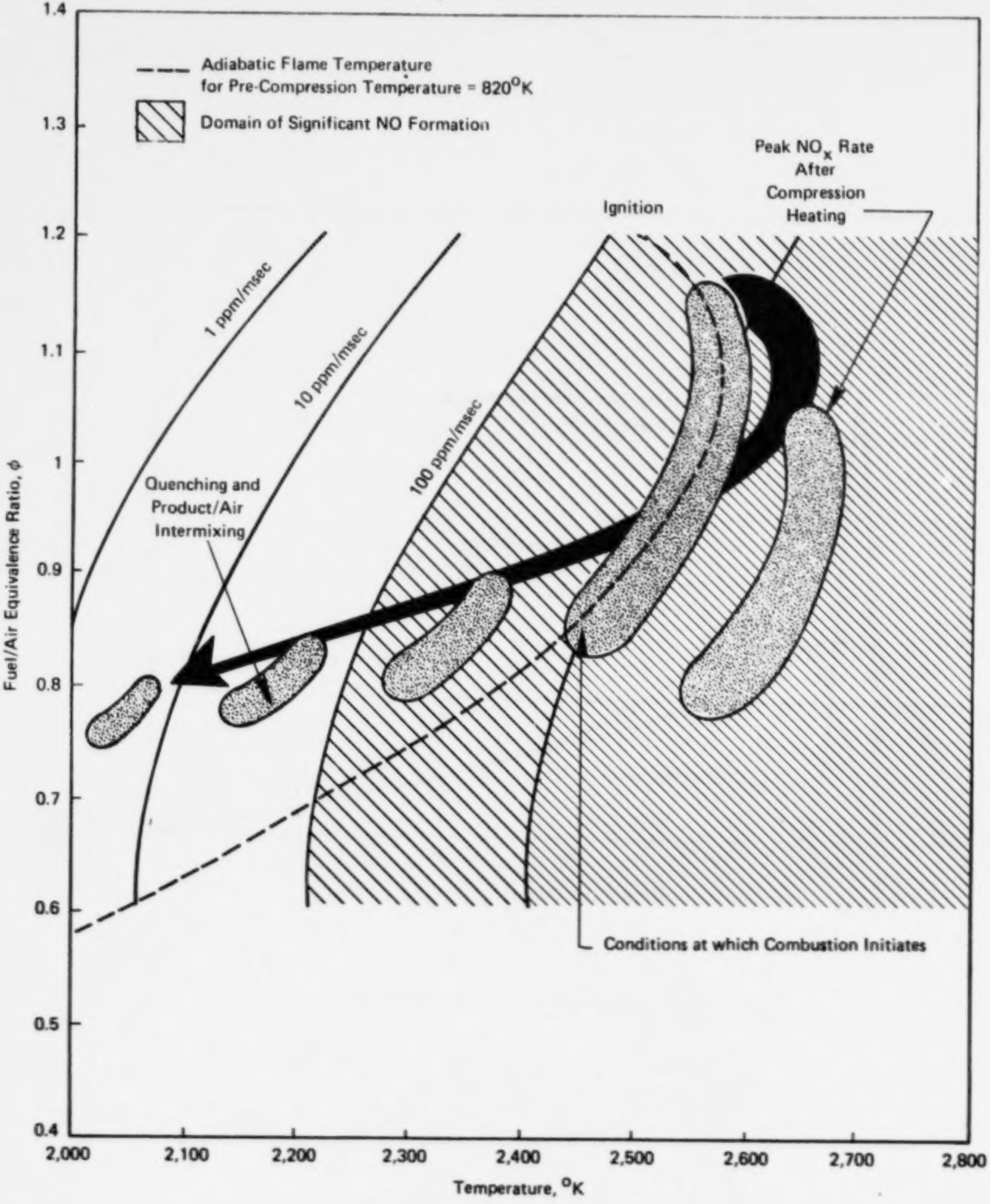
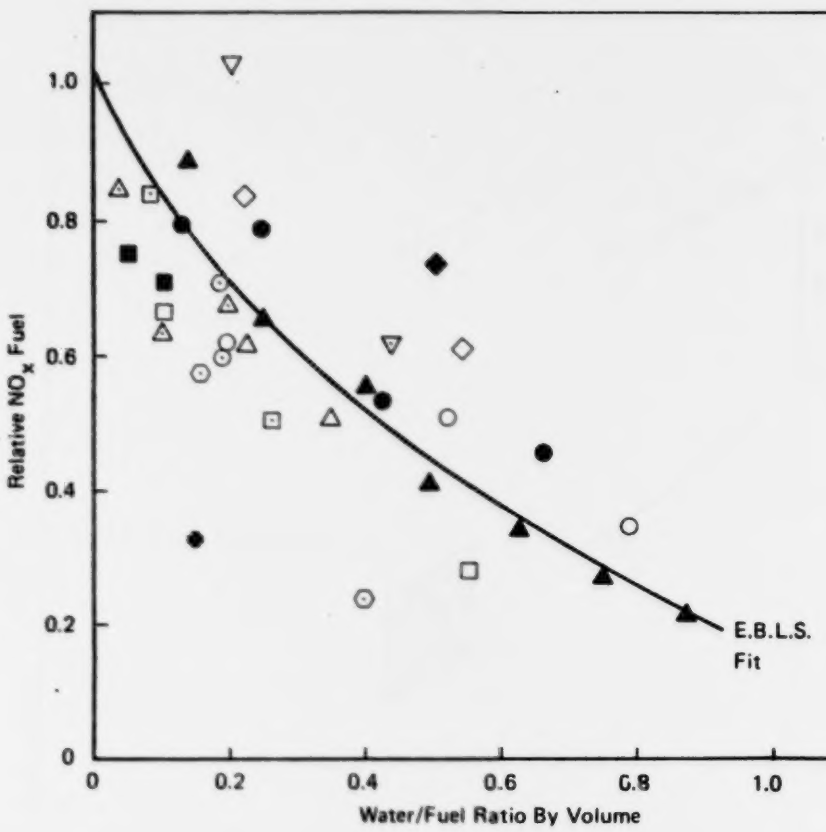
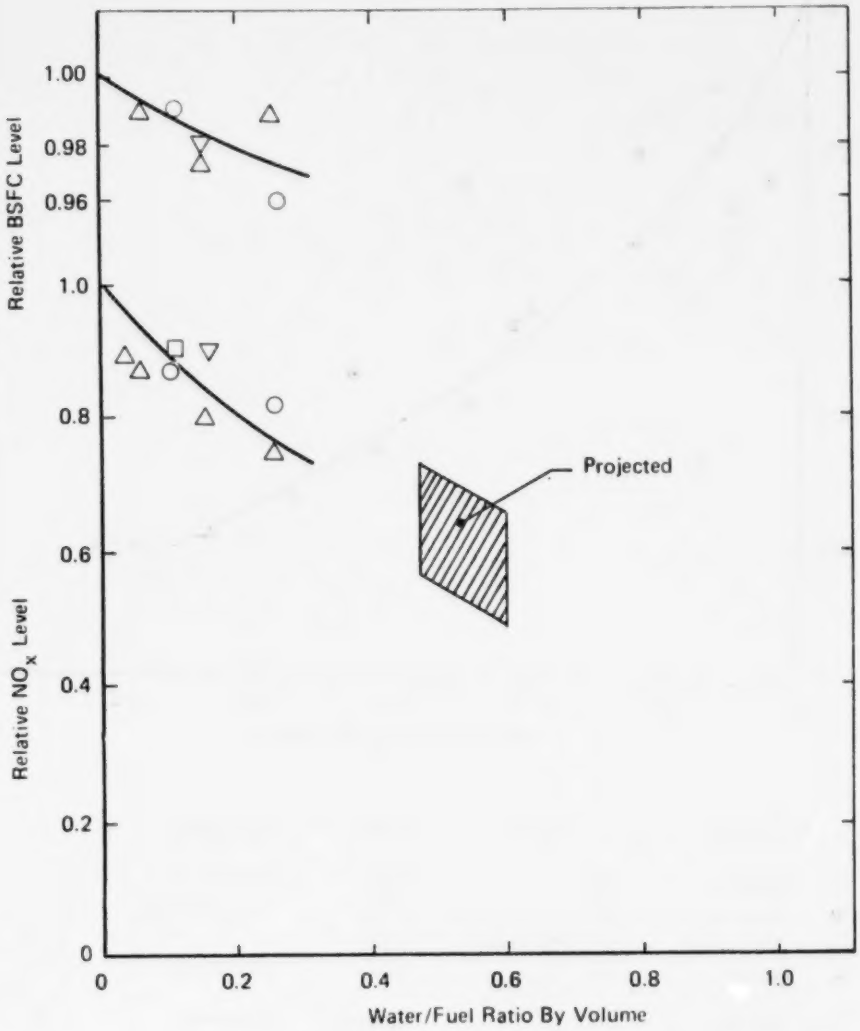


Figure 3–2. Nitric oxide formation rate for a non-uniform fuel distribution. (40 atm pressure)



	First Author	Type	RPM	Emulsifier
○	Greeves	DI	2000	T-Valve
●	Murayama	DI	1200	Twisted Blade
◇	Vichnievsky	DI ("AGROM")	2000	–
▲	Wasser	IDI	1800	Gear Pump
▽	Marshall	DI	13-Mode	Chemical
⊙	Abthoff	DI	2000	–
■	Owens	DI	2300	Chemical
△	Vichnievsky	IDI	2500	–
•	Thompson	DI	1500	–
◆	Valdmanis	DI	2600	–
□	Vichnievsky	DI ("MONO")	2000	–
⊙	Storment	DI	2500	"DA" Type
△	Storment	DI	2000	"DY" Type
▽	Last	DI	1750	Vortex
⊡	Last	IDI	2300	Vortex

FIGURE 3-3: EFFECT OF EMULSIONS ON HIGH-SPEED DIESELS (1200-2600 RPM)



	First Author	Type	RPM	Emulsifier	Bore
○	Taylor	DI (Single)	1000	Gear Pump	216 MM. (8.5")
△	Taylor	DI (Multi)	1000	Gear Pump	~270 MM. (10.5")
▽	Bastenoff	DI (PC–2.5)	500	Westhalea	400 MM. (15.75")
□	Semt	DI (PC–2.5)	500	Homogenizer	400 MM. (15.75")

FIGURE 3-4: EFFECT OF EMULSIONS ON MEDIUM-SPEED DIESELS (400-1000 RPM)

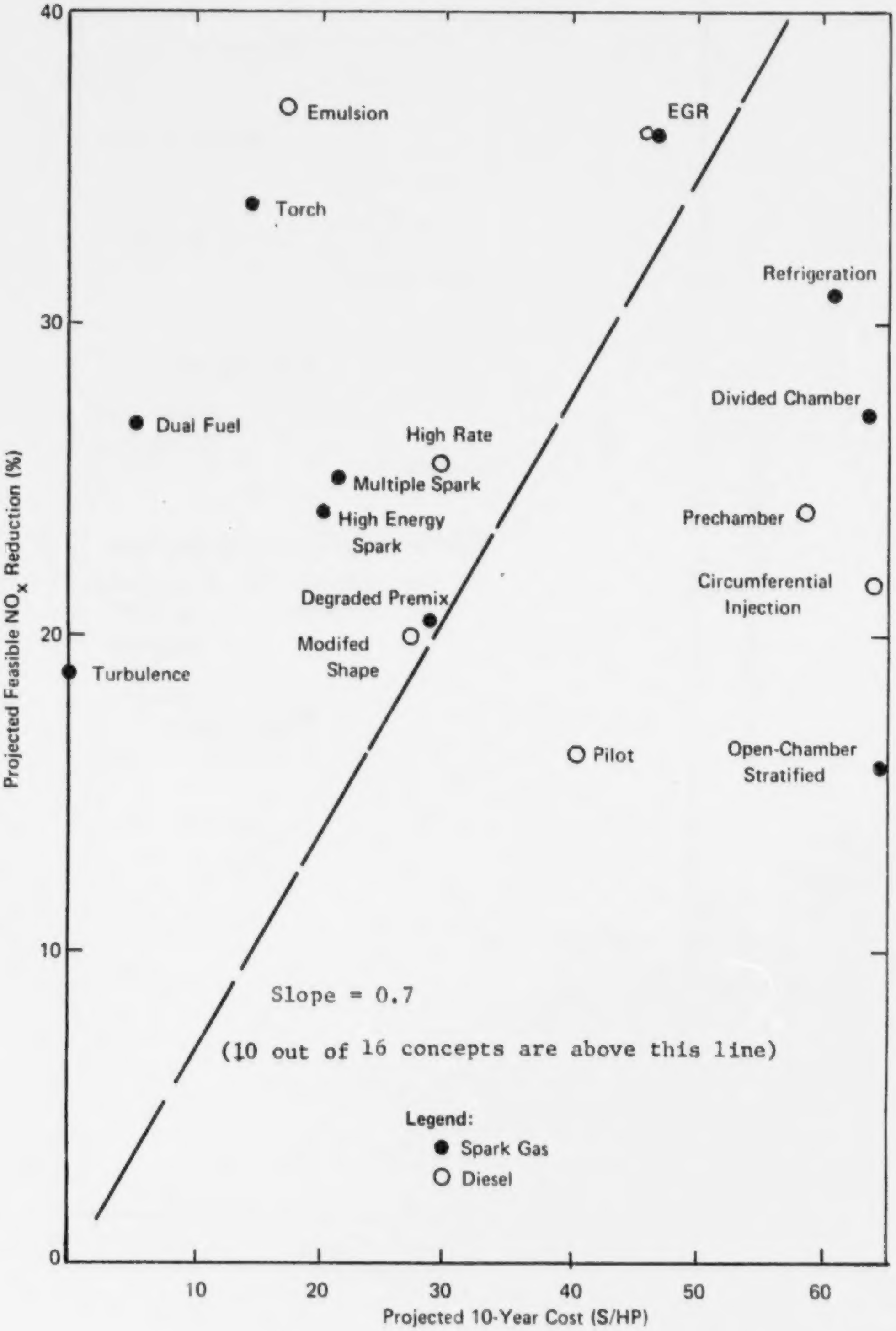


Figure 4–1. **Cost effectiveness of emission control concepts.**

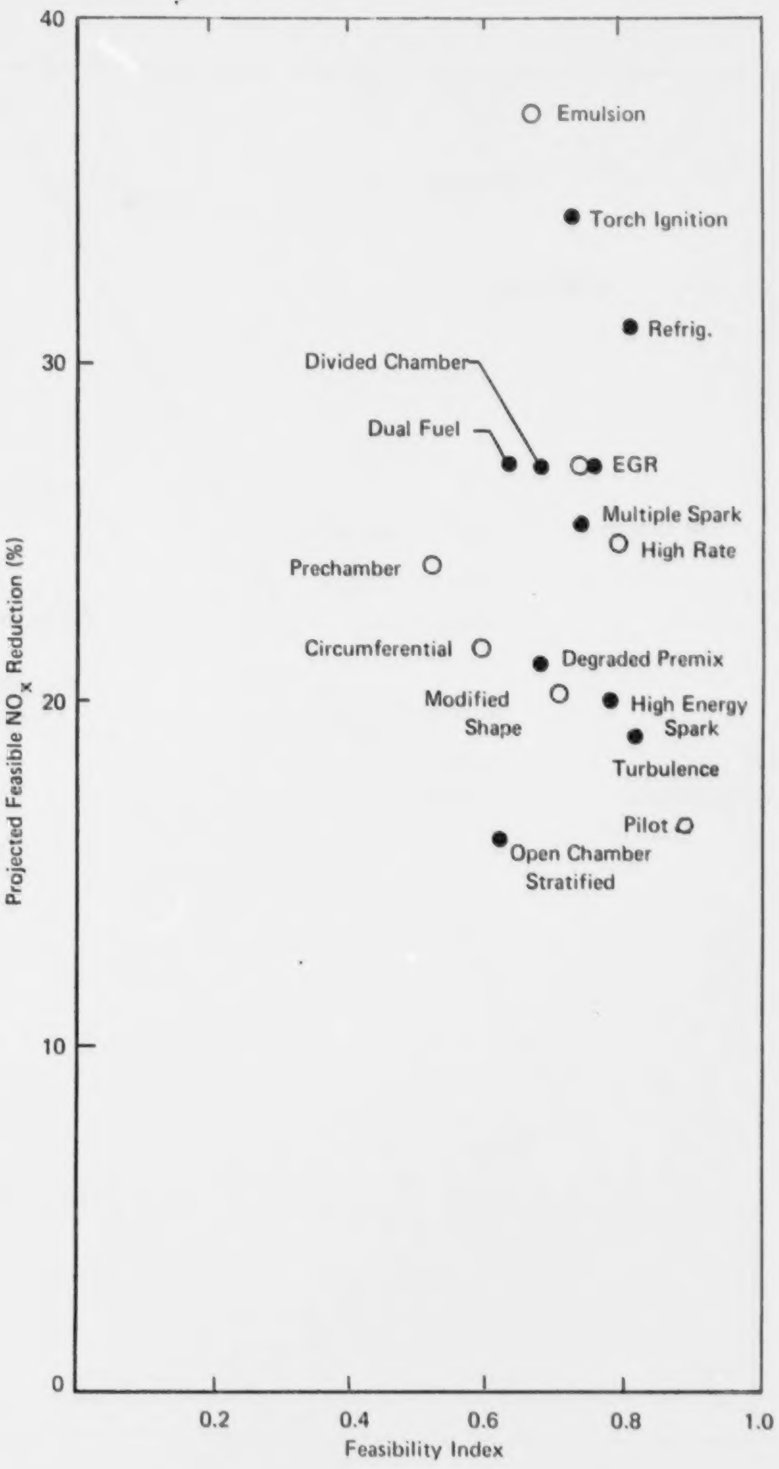


Figure 4–2. **Feasibility and effectiveness of emission control concepts.**

TABLE I. STATIONARY LARGE BORE ENGINE POPULATION

Engine Type	Application	Predominant Engine Size	Estimated Installed Capacity (10^6 HP)	Estimated Annual Fuel Use (10^{14} BTU)	Representative Engine Models
Spark	Gas Pipeline Transmission	2000-4000 HP (14-20" bore)	8.0	3.4	Cooper GMV,GMW Ingersoll Rand KVS Dresser Clark TVC, HBA Worthington UTC
	Gas Gathering, recomprssion, and storage	800-1600 HP (8-10" bore)	4.0	1.7	Superior 510,825 Waukesha VHP
Diesel	Deep oil well drilling rigs and oil transport	1500-2500 HP	6.5	3.0	Waukesha VHP Electromotive 645 Superior 510,825 Fairbanks-Morse 38D
	Baseload electricity generators for municipal utilities	2000-3000 HP (8-10" bore)	4.0	1.7	Same as above
	Standby generating sets for nuclear and hospitals	4000-8000 HP	2.3	0.7	Pielstick PC-2
	Industrial power and water/sewage pumping	1000-2000 HP	1.4	0.6	Same as above
		TOTALS	26.2	11.1	

Source: Arthur D. Little, Inc. estimates based on interviews with engine manufacturers, Youngblood et al(1978), McGowin(1973), Dietzman and Springer(1974), and FPC News, 22 Oct 1976.

TABLE II

Projected NO_x Reductions (Maximum with Under 4% BSFC penalty)

Concept	Based on Engine Data		Based on Model	"Engineering Judgment"	Weighted Consensus[21]	References
	Large Bore	Small Bore				
SI:						
Torch Ignition	35%	50%	28%	30%	34%	1,5
Multiple Spark	--	60	28	22	25	17
Increased Turbulence	--	--	20	18	19	--
Diesel Fuel Ignition	30	--	--	20	27	18
Feedback Control	--	--	--	--	--	--
Degraded Premix	--	--	27	15	21	--
Divided Chamber	--	42	32	23	27	19
Open Chamber Stratified	--	26	19	13	16	20
High Energy Spark	--	50	28	15	20	16
Charge Refrigeration	35	40	30	25	31	2,3
SI or Diesel:						
Timing	15	--	15	15	15	4
EGR	33	--	46	20	27	6
NH_3 w/Catalyst	--	(80%)	--	50	65	7
Ozone w/Scrubber	--	--	--	50	50	--
Nitrogen Plasma	--	--	--	50	50	--
Absorption	--	--	--	--	--	--
Diesel:						
Water/Fuel Emulsion	45	65	--	28	37	8,9
Pilot Injection	15	20	--	17	16	10,11
Prechamber	20	40	--	28	24	11,12
High Injection Rate	29	60	--	22	25	13,14
Modified Chamber	16	50	--	23	20	11,15
Circumferential Injection	--	--	--	22	22	--

[1] McGowin et al(1973)
[2] Blumberg and Kummer(1971)
[3] Wilson et al(1978)
[4] Youngblood et al(1978)
[5] Wyczalek et al(1978) and Sakai et al(1976)
[6] Youngblood et al (1978) engine #70
[7] Pozniak(1975)
[8] See Figure 3-3
[9] See Figure 3-4
[10] Wilson et al(1978)
[11] Hermann(1977)
[12] Wilson et al(1973)
[13] Kasel & Newton(1977)
[14] Khan, Greeves, Wang(1973)
[15] Middlemiss(1978)
[16] Dale et al (1978)
[17] Kuroda(1978)
[18] Schaub & Beightol(1973)
[19] Desoete(1977)
[20] Wall et al(1978)
[21] Weighting Factors from left to right are 50%, 0, 25%, 25%

TABLE III

OVERALL RANKING OF EMISSION CONTROL METHODS

	Method	% NO_x Reduction	Rank (NO_x)	10-Year Added Cost	Rank (cost)	Feasibility Index F	Rank (Feasibility)	Δ BSFC (± 2%)
SPARK IGNITION	Torch Ignition	34%	1	$14/hp	3	.73	5	− 1
	Multiple Spark	25	4	21	4	.74	4	0
	Increased Turbulence	19	7	-3	1	.81	1*	− 1
	Dual Fuel	27	3*	5	2	.65	9	− 2
	Feedback Control	-	-	28	6	.72	6	−
	Open Chamber Stratified	16	8	65	11	.62	10	+ 4
	Degraded Premixing	21	5	29	7	.68	7	+ 1
	Divided Chamber	27	3*	64	10	.67	8	+ 4
	High Energy Spark	20	6	24	5	.79	2	0
	Charge Refrigeration	31	2	61	9	.81	1*	0
	EGR	27	3*	47	8	.75	3	+ 1
	Timing	15	9	−	−	-	-	+ 4
DIESEL	Emulsion	37%	1	$17/hp	1	.68	5	− 2
	Pilot	16	7	42	4	.89	1	+ 2
	Prechamber	24	4	59	6	.56	7	+ 3
	High Rate	25	3	31	3	.80	2	+ 2
	Modified Shape	20	6	29	2	.71	4	+ 2
	Circumferential	22	5	68	7	.60	6	+ 2
	EGR	27	2	47	5	.75	3	+ 1
	Timing	15	8	−	−	-	-	+ 4
EXHAUST TREATMENT	NH_3 Catalyst	65%	1	247	1	.75	3	+ 1
	Ozone	50	2*	934	3	.70	4	+ 4
	Nitrogen Plasma	50	2*	714	2	.80	1	+ 4
	Absorption	-	-	−		.78	2	+ 2

TABLE IV

Ranking of Emission Control Methods based on NO_x Reduction, Feasibility and Cost

	Method	All Factors	
		$\frac{F \cdot \Delta NO_x}{\Delta Cost}$	Rank
SPARK IGNITION	Torch Ignition	1.7	3
	Multiple Spark	0.9	4
	Increased Turbulence	∞	1
	Dual Fuel	3.8	2
	Feedback Control	–	–
	Open Chamber Stratified	0.2	9
	Degraded Premixing	0.5	6*
	Divided Chamber	0.3	8
	High Energy Spark	0.6	5
	Charge Refrigeration	0.4	7
	EGR	0.5	6*
	Timing	–	–
DIESEL	Emulsion	1.5	1
	Pilot	0.3	5
	Prechamber	0.2	6*
	High Rate	0.6	2
	Modified Shape	0.5	3
	Circumferential	0.2	6*
	EGR	0.4	4
	Timing	–	–
EXHAUST TREATMENT	NH_3 Catalyst	0.20	1
	Ozone	0.03	3
	Nitrogen Plasma	0.06	2
	Absorption	–	–

*Tie-score

LOW NO_X COMBUSTOR DEVELOPMENT FOR STATIONARY GAS TURBINE ENGINES

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ABSTRACT

This exploratory development program is being accomplished to identify, evaluate, and demonstrate dry techniques for significantly reducing production of NO_x from thermal and fuel-bound sources in burners of stationary gas turbine engines.

Utilizing the low-NO_x combustor design concept, "Rich Burn/Quick Quench", that was identified previously in the first two phases of this four-phase program, the design of a full-scale prototype combustor has been completed.

The essential features of the NO_x reduction concept, which were determined in early bench-scale experiments, are reviewed and the design of the full-scale combustor is described.

The experimental program now underway to evaluate performance and NO_x-reduction characteristics of the full-scale prototype combustor has progressed through preliminary checkout testing.

INTRODUCTION

The overall objective of the work described in this report is to identify, evaluate, and demonstrate combustion control techniques for significantly reducing the production of oxides of nitrogen in stationary gas turbine engines. While the precise role of the gas turbine engine in future utility and industrial powerplant applications is still emerging, it is clear that the number of units in operation, which has increased dramatically in recent years, will continue to grow. The higher cost of petroleum and the concern over air quality are forcing many sectors of energy industry to re-examine traditional business patterns. Because of the inherent flexibility of installation and operation, short manufacturing lead times, and lower capital costs, the gas turbine engine is projected for use in utility combined cycles and industrial cogeneration applications, as well as the presently used modes. Operation on nitrogen-bound liquid hydrocarbon and low-BTU gaseous fuels, and the capability to meet increasingly stringent exhaust emission requirements are prerequisites to such expanded use.

An exploratory development program was undertaken to achieve significant reductions in both thermal and fuel-bound NO_x in combustors representative of the designs employed in both current and future stationary gas turbine engines. The investigations have addressed dry combustion control techniques and have been directed toward combustor designs that are suitable for use in a 25 megawatt (nominal) stationary gas turbine engine. The program goal for NO_x in combustors fired with gas or oil containing no appreciable bound nitrogen is 50 ppmv (at 15% O_2); for combustors fired with an oil containing up to 0.5% nitrogen (by weight) the goal for NO_x is 100 ppmv (at 15% O_2). The NO_x goals are to be attained while maintaining CO emission levels less than 100 ppmv (at 15% O_2).

The program is being accomplished in four phases and is currently nearing completion. The first phase, which consisted of an analytical investigation of combustion concepts considered to have potential for reducing the production of NO_x, has been completed. In the second phase of work, a number of promising low-NO_x-production concepts were bench-tested to select the best candidate for implementation into the design of a full-scale, 25 megawatt-size, utility gas turbine engine combustor. The results of the program, through Phases I and II, are summarized in reference 1. In Phase III, which has been completed, a full-scale low NO_x combustor was designed and fabricated. In the last phase of work, now in progress, the NO_x-reduction capability of the prototype full-scale combustor is being examined experimentally at conditions simulating the operating range of a 10:1 pressure-ratio stationary engine.

PHASE I AND PHASE II RESULTS

In the first two phases of work, a review and analytical study were conducted to identify concepts that might have potential for reducing the production of NO_x from thermal and fuel-bound sources of nitrogen in stationary gas turbine engine combustors. The most promising of these were selected for experimental evaluation in bench-scale hardware. Of the two successful design concepts that emerged from this experimental study, a concept based on rich burning was selected as the basis for the full-scale combustor design executed in Phase III and evaluated experimentally in Phase IV.

Concept Selection

By general classification, about half the concepts evaluated in the Phase II bench scale program were based on fuel-lean burning, and half were based on fuel-rich burning. Because the conversion of fuel-bound nitrogen to NO_x occurs very readily in fuel-lean combustion, those concepts based on fuel-lean burning were viewed as generally limited to use with fuels containing no appreciable fraction of chemically bound nitrogen. Concepts based on fuel-rich burning, on the other hand were shown to produce low concentrations of the NO_x formed from both fuel nitrogen and atmospheric nitrogen sources. Two of the design concepts examined in Phase II were found to provide substantial reductions in NO_x concentration levels, and were judged to be generally acceptable with regard to the fulfillment of conventional engine-combustor design and operating requirements. One of the two concepts was based on lean burning, the other was based on rich burning.

Following the bench-scale experimental program, the various NO_x control concepts that had been evaluated were reviewed relative to their potential and ultimate suitability for incorporation into a stationary gas turbine engine. Also, an assessment was made of the anticipated differential between the originally stated and attainable program goals. This effort was considered essential to determine if the stated goals were realistic or if they should be revised as dictated by the experimental results obtained in Phase II or by other information obtained in the conduct of the program.

It was concluded that the program goals as originally stated were indeed realistic and that the two successful concepts identified in the bench-scale experimental program were capable of meeting the required NO_x and CO concentration levels. Of the two concepts, the lean-burning approach had been found to meet the emission goals only for fuels containing no more than a trace concentration of chemically bound nitrogen. The rich burning concept showed significant potential and had been demonstrated to meet program goals for NO_x reduction with fuels containing nitrogen in concentrations up to 0.5% by weight.

Based on this assessment, the rich-burning concept was selected for implementation into the design of the full-scale combustor in the final two phases of the program.

For review, the key elements of the rich burning concept are identified in figure 1. A premixing chamber is provided, in which the fuel is prevaporized and premixed with air to form a homogeneous rich mixture. The prepared mixture is introduced into a primary zone section of the combustor and burned without the further addition of airflow. The rich burning process is terminated in a final step involving very rapid dilution, which provides the airflow needed to achieve an overall lean exit plane equivalence ratio. The success of this concept, which does not differ in its essential features from many previous proposals for a rich burn, quick quench approach to NO_x reduction, has been largely a matter of execution and of the selection and refinement of techniques for achieving the idealized conditions called for in the basic concept.

The arrangement of the rich burner bench scale hardware is shown in figure 2. A single, high velocity premixing passage is provided, terminating in a swirler that serves to stabilize the flame in the primary zone of the combustor. All the air entering the primary zone comes through the premixing passage. At design point, the primary zone operates fuel rich. It is followed by a dilution section has has been designed for very rapid quenching of the fuel rich gases leaving the primary zone.

Tests of the rich burner were conducted at elevated pressures and temperatures, simulating actual engine operating conditions. In figure 3 data are shown from tests conducted at 150 psia, at inlet air temperatures of 650°F and 750°F. By staging the amount of air that entered the premixing tube, it was found that low NO_x concentration levels could be achieved over a range of overall (exit-plane) equivalence ratios. At the primary air settings shown (7% and 14%), NO_x concentrations of 60 ppmv and lower were demonstrated using No. 2 fuel with 0.5% nitrogen (as Pyridine). Even lower concentrations were demonstrated using non-nitrogenous fuel. In figure 4, representative bench-scale data points are presented for the rich burning concept, demonstrating low NO_x concentration levels over a wide range of operating conditions, using No. 2 fuel.

Tests of the rich-burning concept were also conducted using a low-BTU gaseous fuel. The exhaust emission data and the composition of the gaseous fuel mixture (synthetically prepared) are shown in figure 5 and figure 6; the arrangement of the bench scale hardware is shown in figure 7. The emission characteristics of the combustor, measured for low BTU gaseous fuel, were very similar to those obtained previously for No. 2 fuel. A minimum NO_x concentration of about 80 ppmv (uncorrected) was measured at the bottom of the "bucket" in the NO_x curve. By varying the primary air setting (which was not attempted in the tests conducted), it should be possible to achieve this same concentration level at any desired operating point over a wide range of overall (exit-plane) equivalence ratios, in keeping with the results of staging tests conducted for the same concept using No. 2 fuel.

PHASE III. FULL-SCALE COMBUSTOR DESIGN

In Phase III the design of a full-scale combustor incorporating the successful NO_x-reduction concept demonstrated in the bench-scale screening experiments of Phase II was carried out. Execution of the design of the full-scale combustor was based in large part upon the data generated in the bench-scale combustor program. It is important to point out, however, that while these results may provide a full-characterization of the bench-scale combustor itself, they cannot be used to specify the complete design of the full-scale combustor. Scaling criteria dictate that there can be no exact and complete correspondence between a prototype combustor and its subscale model, with regard to physical dimensions, operating conditions, and combustion performance. In lieu of direct scaling, a partial modeling approach has been taken, as described in this section. In the basic features of the full-scale combustor, and in the areas of primary air staging (to control stoichiometry), combustor aerodynamics, liner cooling, and residence time requirements, an attempt has been made to reproduce the essential processes of the rich burning concept, as identified and defined parametrically in the bench-scale test results. The design of the full-scale combustor has been executed separately, drawing upon analytical modeling techniques and upon the bench testing of key components (particularly the full-scale premix tube) to verify that the essential processes of the concept have been successfully reproduced.

In the discussions that follow, the rich burning concept will be referred to throughout by the descriptive name "Rich Burn/Quick Quench".

Design Requirements

The objectives adopted for the design of the full-scale prototype combustor reflect many of the requirements of conventional gas turbine combustion systems (temperature rise, pressure drop, and others), as well as the stated emission goals of the current experimental development program. It is intended that the NO_x-reduction technology generated in this program be compatible with current state-of-the-art design practice for stationary gas turbine engines in the 25-megawatt-size range. The design requirements for the full-scale combustor are presented in Table I.

Emission-Reduction Concept (Rich Burn/Quick Quench)

The basic features and demonstrated results (from Phase II bench-scale testing) of the Rich Burn/Quick Quench Concept, in summary form, are as follows:

Arrangement - Two combustion zones are arranged in series: a fuel-rich primary zone and a fuel-lean secondary zone, separated by a necked-down "quick quench" section. A diagram of the bench-scale configuration as tested is shown in figure 2, with the major zones identified.

Critical Features - Three key requirements for low exhaust emissions have been identified, using distillate and low Btu gaseous fuels:

- all air entering the fuel-rich primary zone must be premixed with fuel to prevent diffusion burning (in particular, liner cooling airflow cannot be discharged into the primary combustion region);
- minimum NO_x concentration levels are obtained at primary zone equivalence ratios near 1.3;
- quick-quench air is added at a single site, and must be introduced in a manner that produces vigorous admixing, approximating a step change in composition and temperature.

Emission Characteristics - The emission characteristics, or "signature" of the basic concept are shown in figure 8, as generated at a constant airflow setting by varying the burner fuel flowrate. This "signature" has two notable features:

- a peak in the CO curve, to the right of which (at 0.2 exit plane equivalence ratio and higher) measured concentration levels are low;
- a minimum point or "bucket" in the NO_x curve, which corresponds approximately to a primary zone equivalence ratio of 1.3. The NO_x curve "bucket" represents the unique low-emission design point of the basic emission signature.

Variable Primary Zone Airflow - Variable geometry can be employed to shift the low-emission design point over a broad range of exit plane equivalence ratios, as shown in figure 3. As described in reference 1, the NO_x "bucket" can be shifted in this manner while maintaining an essentially fixed CO characteristic.

Residence Time Requirements - The minimum NO_x concentration levels attained (at the bottom of the NO_x curve "bucket") have been shown to decline with increasing primary zone residence time, and with an increasing level of primary zone turbulence. This characteristic results in basic design tradeoffs among primary zone length (residence time), combustor pressure drop, and resultant NO_x concentration levels.

Basic Features of the Full-Scale Combustor

To initiate the design of the full-scale combustor, studies were conducted to determine what methods might be employed to successfully reproduce the critical features of the bench-scale combustor and accomplish the essential processes of the Rich Burn/Quick Quench concept. As stated, the bench-scale combustor hardware cannot be "scaled-up" directly to produce a full-scale design. However, the parametric data generated for the bench-scale combustor does serve to identify the critical features of the bench-scale design, and to characterize the essential processes of the basic concept. To achieve emission characteristics in the full-scale design comparable to those demonstrated in bench-scale, it is necessary to execute a second design (in larger scale) that successfully sets up the same basic physical processes and preserves the critical features of the smaller combustor.

In the following sections, the basic features of the full-scale configuration are described, and discussions of the various procedures followed in the execution of the detailed design of the combustor are presented.

With reference to figure 9, the basic features of the configuration are as follows:

a. A single centrally mounted premixing tube is provided having a velocity versus length schedule similar to that of the smaller tubes employed successfully throughout the bench scale test program. Variable vanes (not shown) are provided at the premixing tube inlet to regulate the primary zone airflow. The premixing tube is offset slightly with respect to the centerline of the combustor in order to be in-line with an engine diffuser passage.

b. An extended length primary zone is provided for increased residence time.

c. A primary liner cooling scheme is provided that does not call for discharging spent cooling air into the combustion region of the primary zone. Airflow from the primary-liner convective cooling passage is discharged into the combustor through the quick-quench slots.

d. The quick-quench section is designed to provide strong mixing, so that an abrupt termination of the primary-zone rich burning process can be achieved. An area ratio of 2.8 to 1 was adopted in the "necked-down" section of the combustor, matching the optimum value determined in the bench-scale tests.

e. The aft dilution zone of the combustor is combined with the engine transition duct to provide a maximum allocation of the available combustor length for the oxidation of CO while still maintaining an extended-length primary zone in the interest of achieving low NO_x.

In the remaining discussion of the design study that has been carried out to determine the detailed configuration of the full-scale combustor, activities are described in four major areas. In figure 10 these areas are identified, and the logic of the overall design study is depicted.

Primary Air Staging

The bench-scale test results from Phase II have consistently shown that minimum NO_x concentration levels are achieved when the primary zone equivalence ratio is maintained near a value of 1.3. In order to achieve this value over a broad range of combustor exit plane equivalence ratios (engine power settings), a method of varying the amount of airflow admitted to the primary zone is required. At the baseload setting, slightly more than 20% of the total combustor airflow is required in the primary zone; at idle, approximately 10% is required.

The method of primary air staging selected for the full-scale combustor is depicted in figure 11. A variable damper, consisting of two sets of vanes (one movable, one fixed) is mounted at the inlet plane of the premix tube. The variable damper can be adjusted to achieve a 2:1 variation in premix tube airflow. At the full-open setting, only a nominal pressure drop (less than 0.1%) is incurred by airflow passing through the vanes. A large number of narrow vanes is employed, to minimize wake formation in the incoming airflow. In going from the full-open to the full-restricted setting, the total damper travel required is only about 10 degrees (or 0.25 inches at the maximum diameter).

Combustor Aerodynamics

The combustor internal airflow distribution is determined by several factors, which include the relative areas of openings in the combustor liner, the pressure/velocity distribution of the approach airflow, and the combustor internal geometry cross-sectional area as a function of length. The full-scale prototype combustor must meet a prescribed schedule of internal equivalence ratios and therefore must be designed for a specific internal airflow distribution.

The Rich Burn/Quick Quench concept calls for a unique "necked-down" shape that produces locally high velocities in a quick-quench section for the purpose of vigorous mixing. An analysis of the effect of these high velocities on the combustor pressure drop and airflow distribution shows that significant "mixing losses" are incurred in the quick quench section, and that these losses must be considered in tailoring the liner hole pattern to achieve the required airflow splits (these mixing losses are believed to be desirable and, in general, to be indicative of the high rate of mixing achieved in that section of the combustor).

To ensure an accurate determination of the liner hole areas required in the full-scale prototype combustor, a computer model was formulated to simulate the aerodynamic processes described above. The model accepts as input a prescribed fractional airflow distribution, the inlet air temperature and pressure, the fuel flowrate, and the required liner pressure drop. The cross-sectional area profile of the combustor is also input, and an external pressure distribution may be specified. The calculation is performed in a downstream-marching fashion, beginning with an initial guess for the premix tube airflow in pounds per second. At each of several stations along the length of the burner, the pressure drops associated with various components and processes are computed. These pressure drops include the following: 1) premix tube entrance and blockage losses (both at the variable damper and at the fuel injector); 2) swirler pressure loss; 3) momentum pressure loss; 4) mixing loss in the quick quench section; 5) mixing loss in the dilution zone. At the exit plane a check is made on the overall pressure drop. If it agrees with the specified input value, the solution is complete. Otherwise a new value for the premix tube airflow rate is assumed, and the computation is repeated. The final solution includes the total airflow that can be passed through the combustor for a given overall pressure drop and specified distribution, and the schedule of hole areas required to achieve that distribution.

Several cases were run with the aerodynamic model for the purpose of sizing the holes in the quick quench section of the combustor and in the dilution zone. The results verified that a major source of combustor pressure drop is the "mixing loss" in the quick quench section. The model assumes one-dimensional flow and computes as "mixing loss" the total pressure drop due to mass addition (from the momentum equation). In the quick-quench section, the mass added through the penetration holes is assumed to have zero axial velocity. This flow must be accelerated, along with the approach flow from the primary zone, to a uniform axial velocity consistent with the cross-sectional area of the "necked-down" (quick quench) section of the burner. The smaller the diameter of the "necked-down" section, the greater the required acceleration, and the greater the resultant total pressure drop.

To illustrate the results described, a representative case run with the aerodynamic model is presented in Table II. Predictions for the prototype combustor operating at 5.5 percent pressure drop and at a baseload power setting are shown. The data include computed flow properties at selected stations along the length of the combustor. The stations are identified in figure 12. It may be seen from the tables that there is a progressive decline in total pressure, caused by the losses incurred at the various stations. A major source of pressure drop and a controlling factor in the predicted aerodynamic characteristics of the combustor is the loss incurred in the quick quench section. The quick quench section has a throttling effect on the combustor flowrate. The higher the axial velocity in the necked down passage, (i.e., the smaller the diameter, for a given primary air setting) the lower the quantity of airflow (the total of primary air and quick quench air) that can pass through that section without an increase in burner pressure drop.

These predicted results have been verified experimentally in tests of the bench-scale combustor, as shown in figure 13. Good agreement with the experimental data was demonstrated.

Liner Cooling Scheme

A critical feature of the Rich Burn/Quick Quench concept is the elimination of non-premixed air from the primary zone of the combustor. The primary liner cooling scheme depicted in figure 9 calls for convective cooling of the outside surface of the liner, and for the discharge of spent cooling air into the quick quench section of the combustor. Cast fins are provided on the cooled side of the liner to increase the effective surface area. An outer shroud is placed around the cast liner to maintain a high air velocity along the outside surface.

To ensure that the intended design can be properly implemented, and that the required cooling effectiveness can be achieved, an analysis of the aerodynamic and heat transfer characteristics of the convective cooling channel was conducted. Model predictions for the maximum heat-load condition (which occurs at an equivalence ratio of about 1.2) indicated that a peak primary liner temperature of 1536°F can be expected when 43.2% of the total burner airflow is used for cooling. This relatively high percentage of the total airflow is readily available for cooling because it also serves as quick quench air.

To verify the results of the analytical studies and to assess the effectiveness of the convective cooling technique, a short series of bench scale experiments was also carried out. The data generated in these experiments (using reduced scale hardware) were used as a standard of validation for the analytical model predictions with regard to the influence of burner airflow rate, inlet pressure, and inlet air temperature on the primary liner wall temperature level.

The predicted effect of inlet temperature on wall temperature is shown in figure 14. The agreement between the model and data is very close. An increase in inlet temperature from 400°F to 600°F roughly increases the maximum wall temperature (at an overall FA of 0.070) from 1300°F to 1600°F.

Premix Tube

Good fuel preparation (effective prevaporization and premixing) is of paramount importance in the design of the full-scale combustor. If the airflow entering the primary zone has not been admixed with fuel to form a homogeneous mixture, diffusion burning will take place between the incoming air and the fuel-rich gases already present. Because diffusion burning proceeds at near-peak flame temperatures, it is inevitable that significant concentration levels of NO_x will be formed in the primary zone under these circumstances.

In order to provide uniform premixing (and prevaporization) of the fuel and air that are introduced into the primary zone, a number of candidate designs for the full-scale premix tube have been proposed and evaluated (both analytically and experimentally) during the Phase III design effort. In the course of these evaluations a considerable body of design data has been gathered. These data have been assembled to form a premix tube design system. In this section a brief description of the design system is presented.

At the present time several of the premix tube designs described in the following discussions are principal candidates for evaluation in tests of the full-scale combustor.

a. Atomization

Atomization of the liquid fuel and optimization of droplet sizes is of paramount importance to the designer for two reasons. First, fuel vaporization is dependent on fuel drop size: the smaller the fuel droplet, the faster it vaporizes. Because vaporization is usually one of the attainable goals of a premix system, atomization determines the premixing length requirements for vaporization. Second, even if complete vaporization is not accomplished, small droplets ($\leq 20 \mu m$) can be expected to behave like vapor in the combustion process. Thus small premixed fuel droplets in air can approach the performance of a perfectly premixed, prevaporized system.

Of the various atomization processes, air atomization has the most potential to produce fine droplets in premix tubes. In order to optimize air atomization, three types of fuel injection or combinations thereof can be used:

1) downstream axial injection (low fuel velocities)
2) upstream axial injection
3) cross-stream (radial or tangential) fuel injection

All these types of injection provide a high relative velocity between the fuel and air, thus promoting good atomization.

Empirical correlations for drop size (resulting from air atomization) must be a function of the following parameters:

ν_f - viscosity of fuel

σ_f - surface tension of fuel

ρ_f - density of fuel

ρ_a - density of air

V_a - velocity of air (relative to fuel)

d_f - characteristic initial dimension of fuel (diameter, thickness, etc.)

$\frac{W_a}{W_f}$ - airflow-to-fuel flow ratio.

All other parameters have been shown to have a negligible influence on the Sauter Mean Diameter (SMD).

The last parameter, W_a/W_f, is a droplet interference and interaction term that can be eliminated from the list by the following reasoning. If all of the airflow passing through the premix tube is used in the atomization process, the air-to-fuel ratio can range from about 10 in fuel-rich premix tubes ($\phi = 1.3$) to about 20 in fuel-lean premix tubes. It has been shown (reference 2) that for values greater than five the air-to-fuel ratio does not play a significant role in the atomization process. Thus we have eliminated the term W_a/W_f from further consideration.

The empirical correlation that follows was derived from references 2-11 which include theoretical analyses and experimental data for liquid jets, sheets and droplets. The correlation has the form

$$\mathrm{SMD} = K(d_f)^a(\nu_f)^b(\sigma_f)^c(\rho_f)^d(\rho_a)^e(V_a)^f$$

where K, a, b, c, d, e, and f are constants. Table III gives a list of the exponents a thru f from the various references. In reviewing the references, it was apparent that some of the constants were remarkably consistent (particularly b, c, and f) while others varied. By the use of dimensional analysis, three exponents can be calculated from three selected exponents. The following equation was derived:

$$(1) \quad SMD = K(d_f)^{.375}(\nu_f)^{.25}(\sigma_f)^{.375}(\rho_f)^{-.125}(\rho_a)^{-.5}(V_a)^{-1.0}$$

The proportionality constant K was determined to be 48 in reference 7.

Equation (1) allows the designer to predict actual SMD values provided the value of d_f is known. Also, equation (1) allows the designer the capability of providing the effect of changing pertinent parameters. It should be noted that air velocity is the single most important parameter in the atomization of a liquid fuel. As a typical reference, an air velocity of 400 f/s at ambient conditions will shatter a thin kerosene jet (.062 in) into droplets with a SMD of 16 μm .

b. Distribution

In addition to atomization, the proper distribution of fuel in a premix tube must be achieved. Poor fuel distribution results in incomplete atomization due to droplet interaction effects, slower vaporization, and mixture nonuniformity. If a premix system is properly optimized, the fuel must be uniformly distributed throughout the airstream by the time the mixture enters the main combustor.

In bench-scale premix tubes (1 inch diameter), experience has shown that centrally mounted pressure atomizing fuel nozzles are capable of properly distributing the fuel. In larger (full-scale) premix tubes, two techniques appear to offer greater potential for a uniform fuel distribution. First, a centrally mounted injector can be used in combination with an inlet-plane mixing device such as a swirler. An example of this type of fuel distribution system is shown in figure 15. The swirling airstream either centrifuges larger droplets outboard or transports smaller droplets by turbulence. Extreme care must be exercised in the design of this type of distribution system both in the avoidance of reverse flow zones and the avoidance of excessive wall wetting by the fuel. Second, multiple injection sources can be used with or without mixing devices. Figure 16 shows two types of premix tubes using multiple injectors, one with and one without a mixing device.

Multiple radial fuel injectors mounted on the wall of the premix tube can also be employed. This approach offers the advantage of providing a uniform fuel distribution without the complexity of a mixing device. Radial injection also eliminates all internal blockage and provides a "clean" premix tube design. However, the provisions for fuel penetration must be carefully determined to properly distribute the fuel without excessive wall wetting. Designs of this type can be undertaken using the three penetration design curves for radial fuel injection from references 12, 13 and 14. These are shown to be in fairly good agreement in figure 17. Data from reference 4 are also plotted in figure 17.

Another promising candidate for optimum fuel distribution is the radial "spoke" design shown in figure 18. Each spoke has multiple orifice injectors which tangentially feed the fuel into the airstream. The injection system shown has 12 spokes and 36 individual orifices spaced on an equal area basis. Reference 15 employed a similar fuel injection system and obtained excellent premixing results.

c. Pressure Loss

In order to design a premix tube that passes the desired airflow and meets the requirement for overall combustor pressure drop, an assessment must be made of the pressure losses of the various parts of the premix tube itself. As an example, three types of pressure loss occur in the premix tube shown in figure 18: internal blockage loss, diffuser boundary layer loss, and swirler dump loss.

Internal blockage loss is calculated from the one dimensional momentum equation. Diffuser boundary layer loss can be calculated from numerous diffuser pressure recovery maps found in the literature. Swirler dump loss is ordinarily calculated from the one dimensional equation of motion assuming a one dynamic head loss based on the discharge area of the swirler. By summing the losses of the various components and iterating to a specific overall loss, the required "size" of the premix tube can be determined.

Full Residence Time (FRT) and Engine-Compatible Designs

The very low NO_x concentration levels achieved under the Rich Burn/Quick Quench concept were shown in the bench-scale experimental program to depend upon an extended residence time in the fuel-rich primary zone section of the combustor. By varying the diameter and the length of the primary zone in a series of bench scale tests, data were obtained that indicate a trade-off between attainable NO_x concentration levels and the primary zone residence time. These results were utilized in the design of the full-scale combustor.

Bench-scale combustor configurations similar to the one in figure 13 were used to generate residence-time data. Primary zone diameters of three, five, and six inches are illustrated. Two lengths were tested, 9 and 18 inches (measured from the premix tube swirler to the centerline of the quick-quench slots), and in one configuration an enlarged premixing tube (designed to pass 70 percent more airflow) was evaluated. The results obtained are presented in figure 19 in terms of the trade-off between the minimum attainable NO_x concentration* levels and the primary zone residence time. Normalized values of these two parameters are shown, because a direct application of raw bench-scale data in the design of the full-scale combustor is not considered appropriate. Dissimilarities in the two combustors (particularly a difference in surface-to-volume ratio of about two-to-one) preclude the possibility of an exact correspondence between the NO_x concentration levels measured in the bench-scale combustor (for a given value of the bulk residence time) and those that can be expected in the full-scale configuration.

In the design of the full-scale combustor, the general relationship between residence time and NO_x concentration levels shown in figure 19 was assumed. It was also assumed that the absolute levels demonstrated in the bench-scale combustor (50 to 60 ppmv over a broad range of operating conditions, as illustrated in figure 4) could be achieved in the full-scale combustor as well. To select a design-point value of the primary-zone residence time, several factors were considered:

a. To provide an absolute value of residence time equal to that which had been utilized in the bench-scale combustor, a primary zone length about 2.5 times greater than the nominal length available in a representative 25 megawatt engine combustor would be required.

b. Primarily because of the lower surface to volume ratio, it was reasoned that a lower value of residence time might be required in the full-scale combustor. For the initial configuration, a value equal to half the residence time utilized in the bench-scale combustor was selected as generally acceptable.

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* The "minimum attainable NO_x concentration" is measured at the bottom of the "bucket" in the characteristic NO_x curve of the Rich Burn/Quick Quench concept, illustrated in figure 8.

c. Because more than one value of primary residence time is required to establish whether the data being obtained fall on the negative-slope portion or the flat portion of the curve in figure 19, it was decided that two configurations of the full-scale combustor, differing in primary zone length, should be tested.

Based on the above considerations, two configurations of the full-scale combustor were designed. The first, which is depicted in basic form in figure 9, provides a primary zone residence time about half as great as that utilized in the bench-scale combustor. The second, shown in figure 20 represents a lower value of primary zone residence time (intended to provide a second data point). Because of the greater residence time it provides, the first configuration has been designated as the Full-Residence-Time (FRT) version of the full-scale combustor. The second configuration, which meets the basic length requirements of a representative 25 megawatt engine, has been designated the Engine-Compatible version of the full-scale combustor.

The construction of the full-scale combustor hardware was completed under Phase III. A photograph of the FRT combustor during construction is shown in figure 21. The premixing tube and primary liner shroud were not attached in this figure. The fully-assembled configuration is shown in figure 22, except for the premixing tube damper mechanism. A view of the damper is shown in figure 11. In the course of the test program, it is planned that the FRT configuration will be modified to produce the short-length Engine-Compatible design.

PHASE IV - VERIFICATION TESTING

In Phase IV, which is now underway, the experimental evaluation of the full-scale combustor will be accomplished. Both the FRT and Engine Compatible configurations will be tested over a range of conditions spanning the operating requirements of a commercially available 25 megawatt stationary gas turbine entine. Three fuels will be used in the test program: No. 2 distillate; No. 2 with 0.5% N (as pyridine); and a distillate cut shale oil. The first tests are being conducted to provide preliminary verification that the design concept has been successfully transferred from bench-scale to full-scale hardware. Initial tests are being conducted at intermediate pressure using all three fuels.

Initial test results have indicated that the basic emission signature of full-scale combustor is the same as that associated with the bench-scale configuration (figure 8). Although NO_x concentration levels measured in the early tests have not been as low as those achieved with the bench-scale combustor, values lower than the program goal have been demonstrated. At the same time, however, the premixing performance has been inadequate (poor distribution of the fuel and minor damage to the premix tube swirl vanes as a result of preignition). Currently, revisions are being made to the design of the full-scale premixing tube. The initial configuration (the basic arrangement is shown in figure 18), has been replaced by the design shown in figure 20. The better fuel preparation characteristics (improved atomization and distribution), and higher internal velocities provided by this design are viewed as critical elements in the attainment of the full NO_x reduction potential of the design concept.

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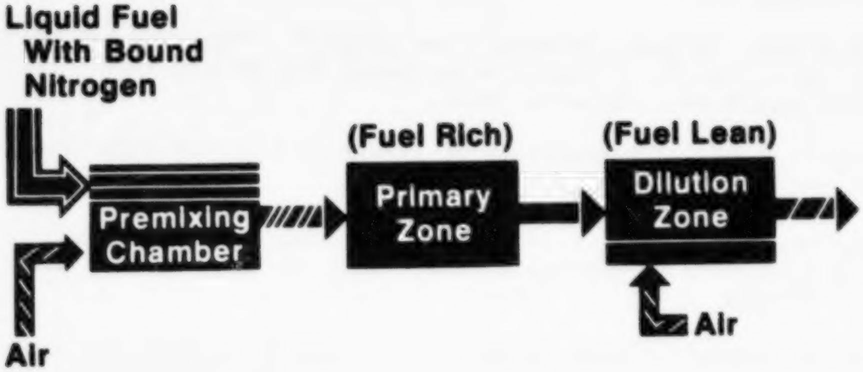

Liquid Fuel
With Bound
Nitrogen
Premixing
Chamber
Air
(Fuel Rich)
Primary
Zone
(Fuel Lean)
Dilution
Zone
Air

Figure 1. Rich Burning Concept Burner Components

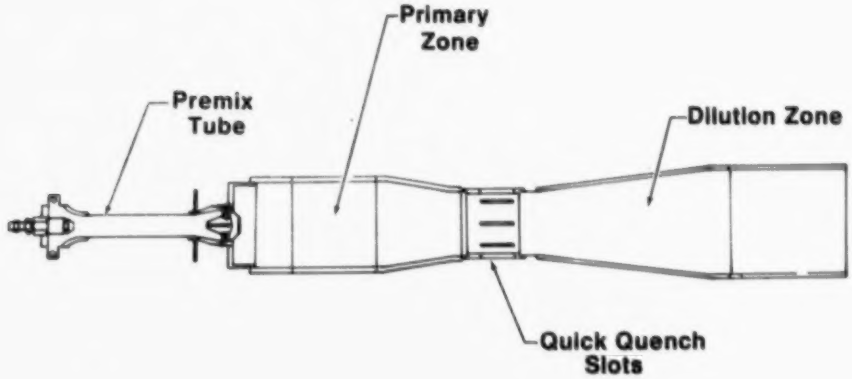

Premix
Tube
Primary
Zone
Dilution Zone
Quick Quench
Slots

Figure 2. Rich Burner Arrangement

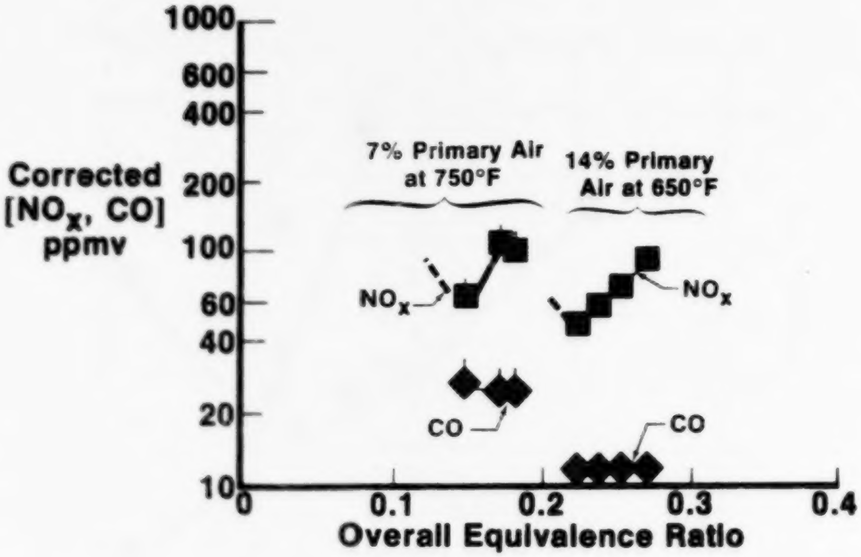


Figure 3. Rich Burner Simulated Engine Cycle Characteristics (150 psia, 0.5% Nitrogen)

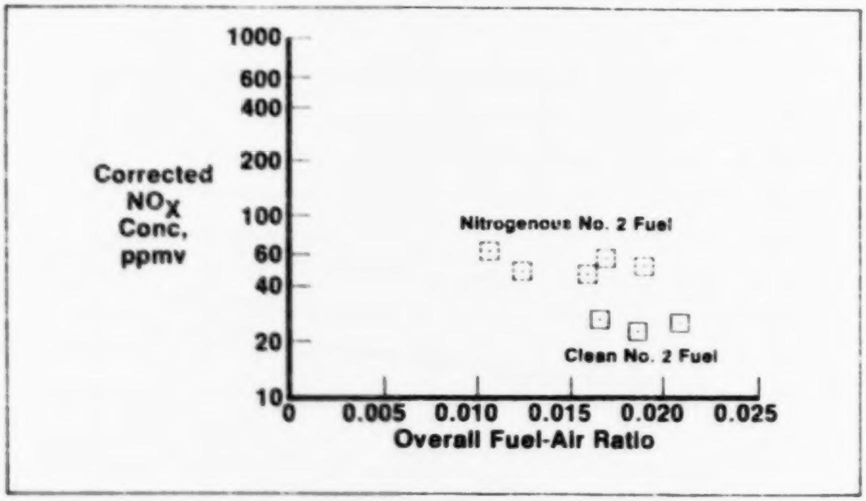


Figure 4. Rich Burn/Quick Quench Combustor Emission Trends

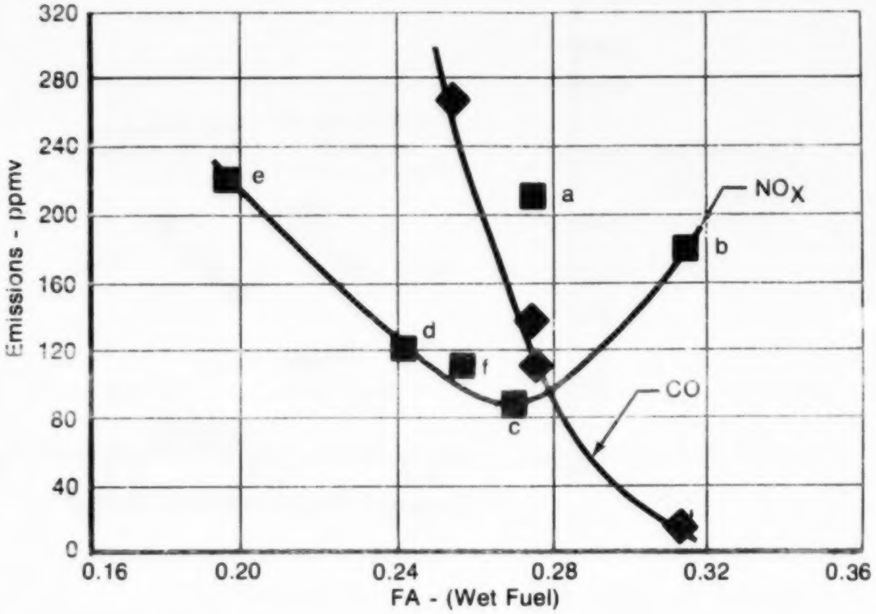


Figure 5. Variation in Emission Concentrations with Fuel-Air Ratio for Tests Conducted with Low BTU Gaseous Fuel

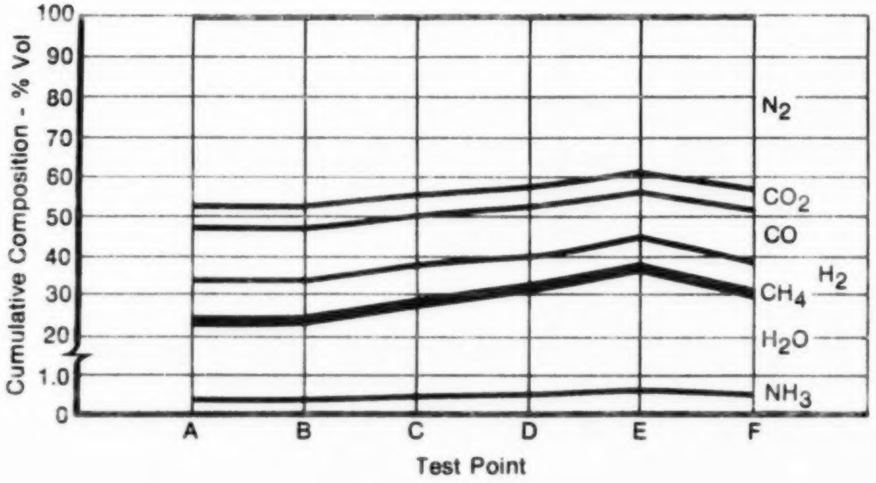


Figure 6. Variation in Low BTU Gaseous Fuel Composition with Test Point Number

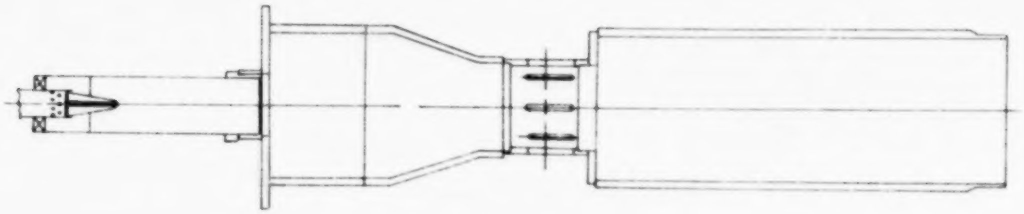


Figure 7. Rich Burner Arrangement for Low BTU Gaseous Fuel Test

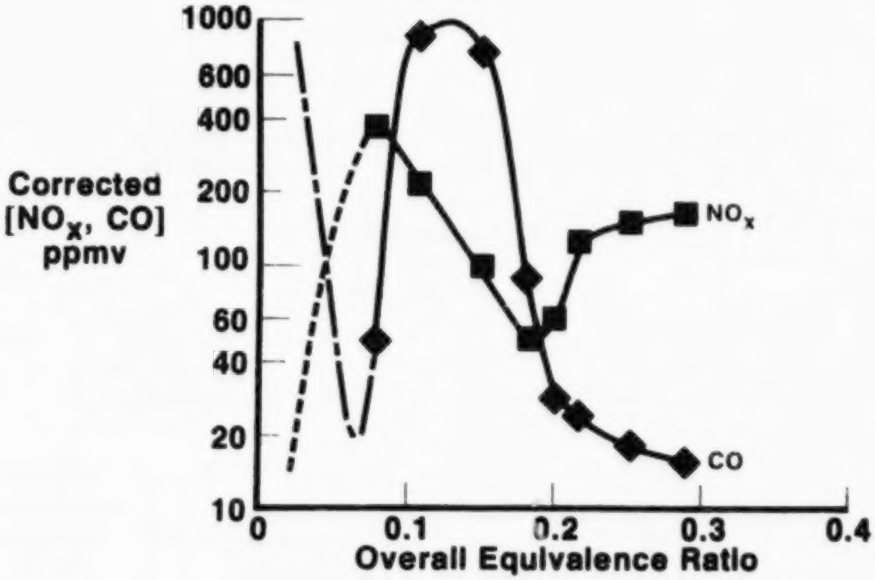


Figure 8. Rich Burner Characteristics (50 psia, 600°F, 0.5% Nitrogen)

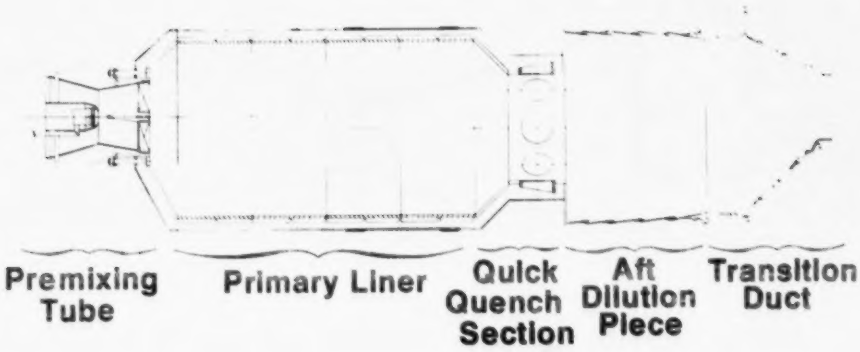

Premixing Tube
Primary Liner
Quick Quench Section
Aft Dilution Piece
Transition Duct

Figure 9. Basic Layout of the Full-Scale Combustor

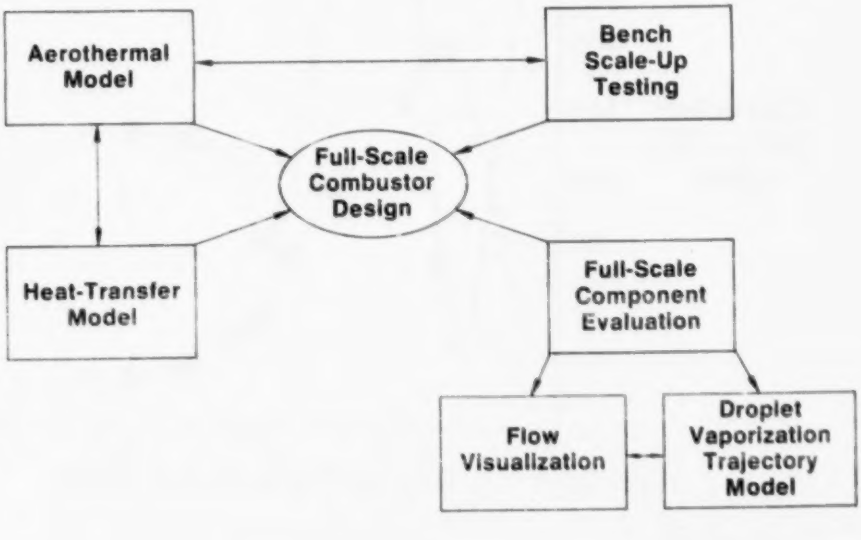

Aerothermal Model
Bench Scale-Up Testing
Full-Scale Combustor Design
Heat-Transfer Model
Full-Scale Component Evaluation
Flow Visualization
Droplet Vaporization Trajectory Model

Figure 10. Major Elements of the Full-Scale Combustor Design Process

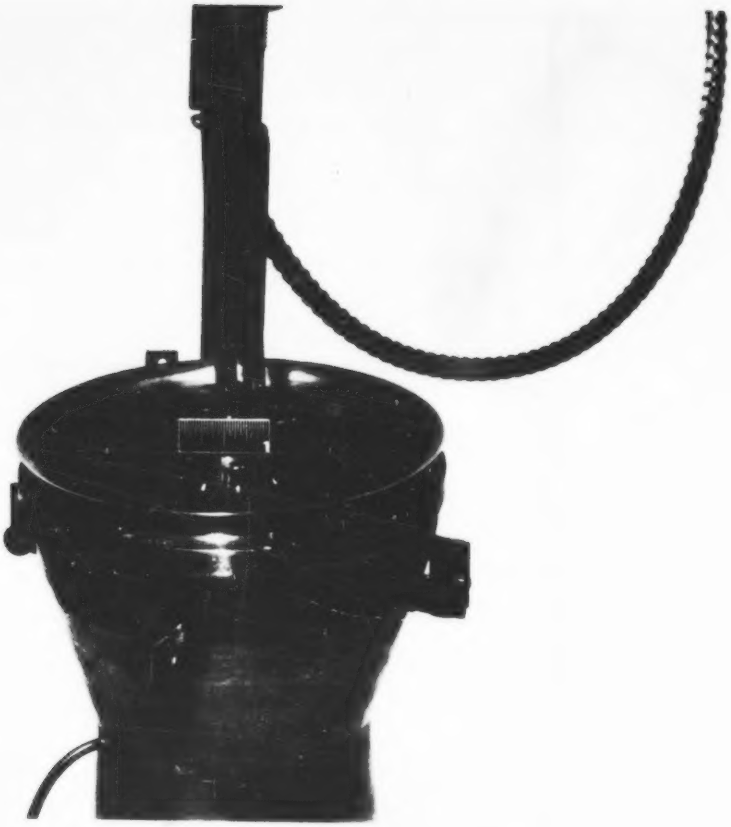


Figure 11. Premix Tube Variable Damper Mechanism

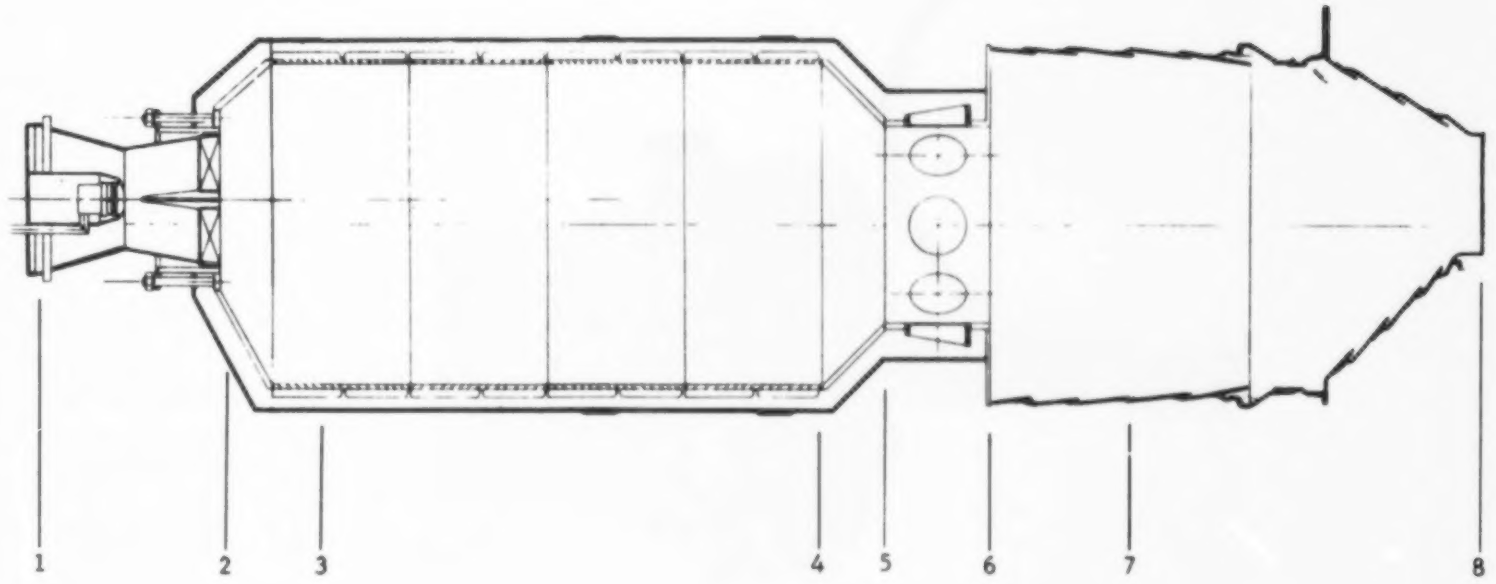


Figure 12. Identification of Stations Referred to in the Aerodynamic Model Calculations

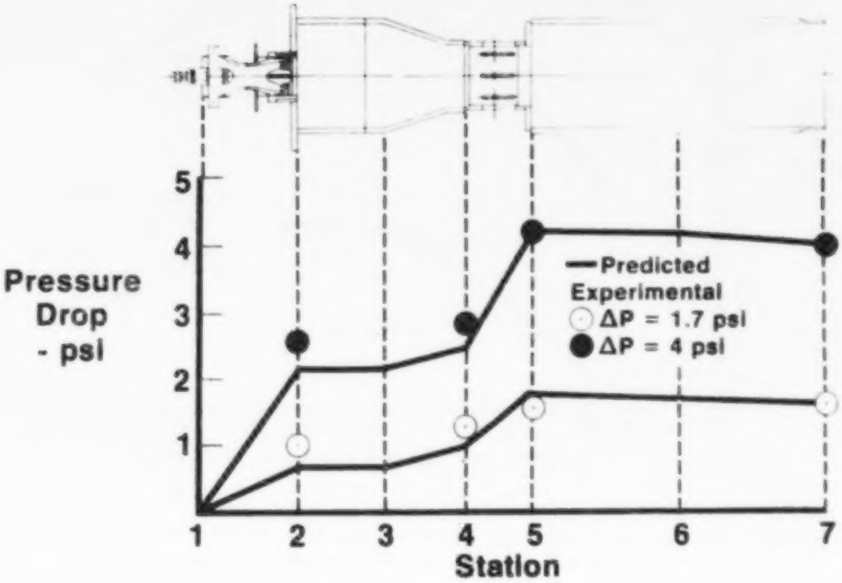


Figure 13. Comparison of Predicted and Experimental Pressure Drop Characteristics of the Bench-Scale Combustor

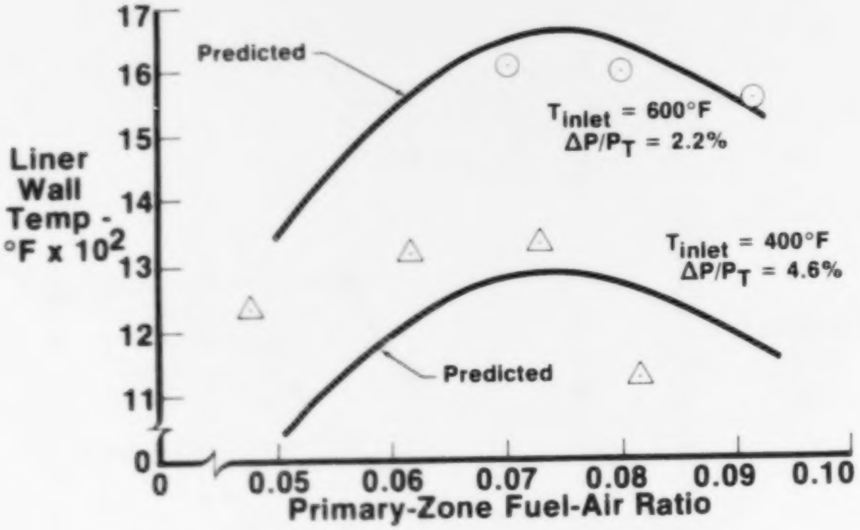


Figure 14. Comparison of Predicted and Experimental Results for the Heat-Transfer Model

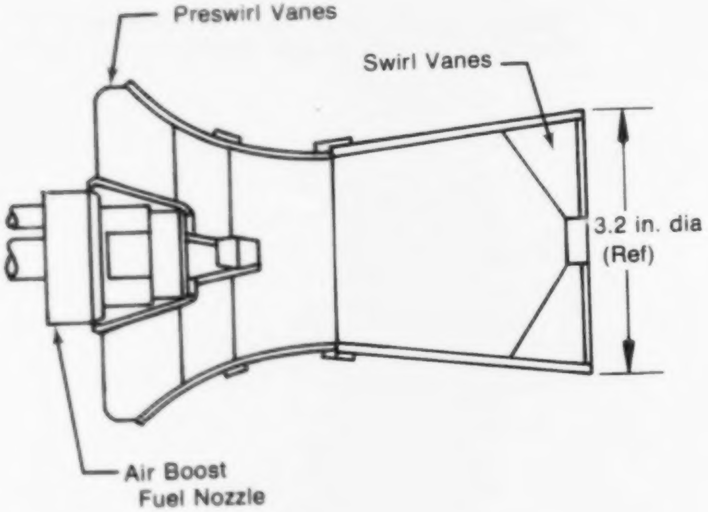
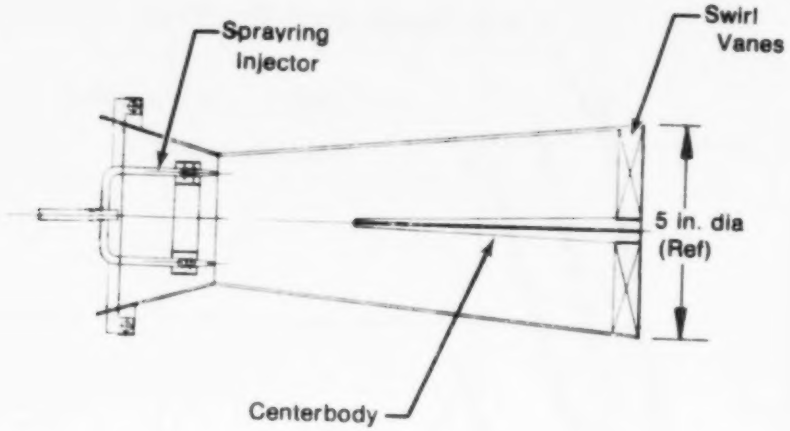
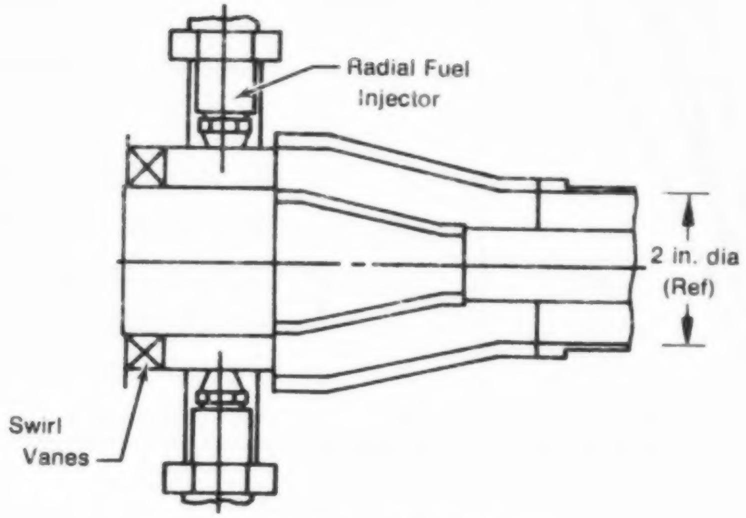

Preswirl Vanes
Swirl Vanes
3.2 in. dia
(Ref)
Air Boost
Fuel Nozzle

Figure 15. A Typical Centrally Mounted Fuel Injector with a Mixing Device


Sprayring Injector
Swirl Vanes
5 in. dia (Ref)
Centerbody

(a) Without Mixing Device


Radial Fuel Injector
2 in. dia (Ref)
Swirl Vanes

(b) With Mixing Device

Figure 16. Typical Multiple Fuel Injector Premix Tubes

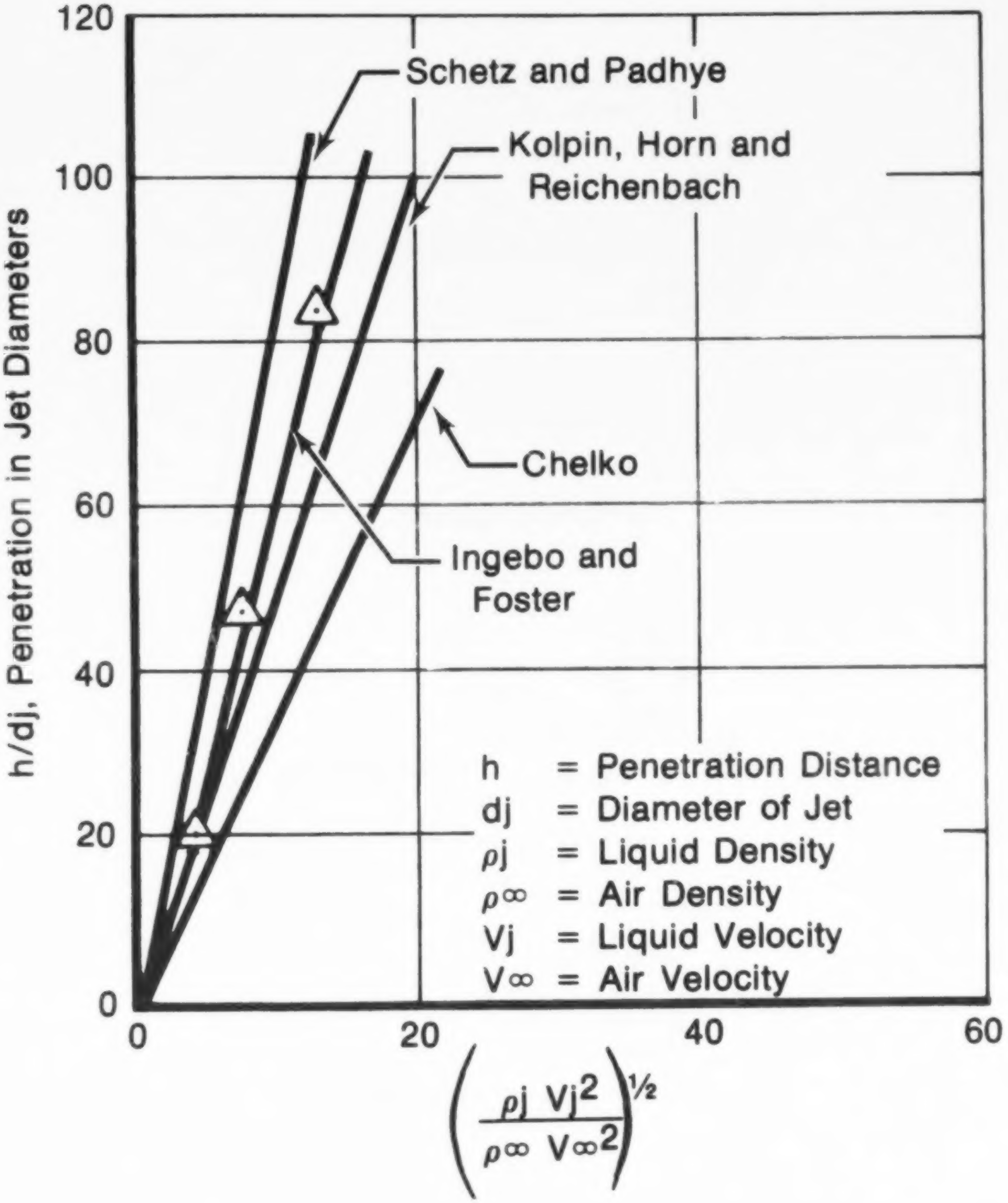


Figure 17. Liquid Jet Penetration in Airstream

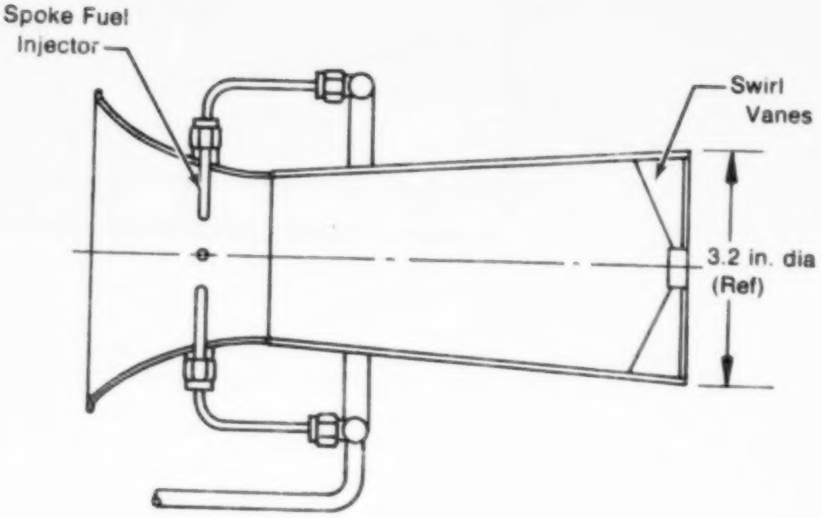

Spoke Fuel
Injector
Swirl
Vanes
3.2 in. dia
(Ref)

Figure 18. Radial "Spoke" Premix Tube Design

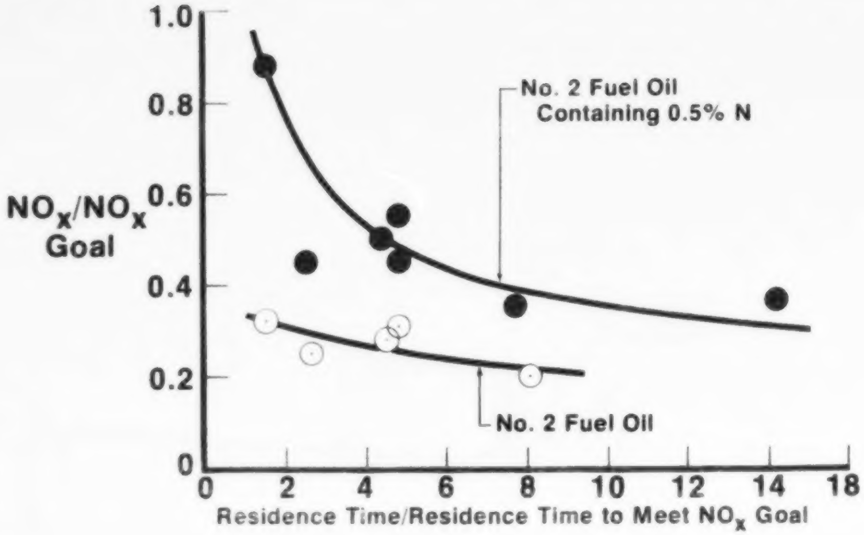


Figure 19. Variation in Minimum NO_x Concentration with Primary Zone Residence Time (Normalized Bench-Scale Data)

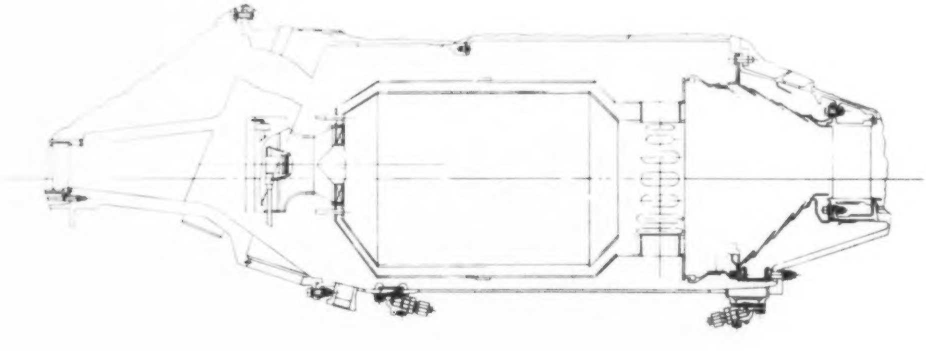


Figure 20. Engine-Compatible Version of the Full-Scale Combustor

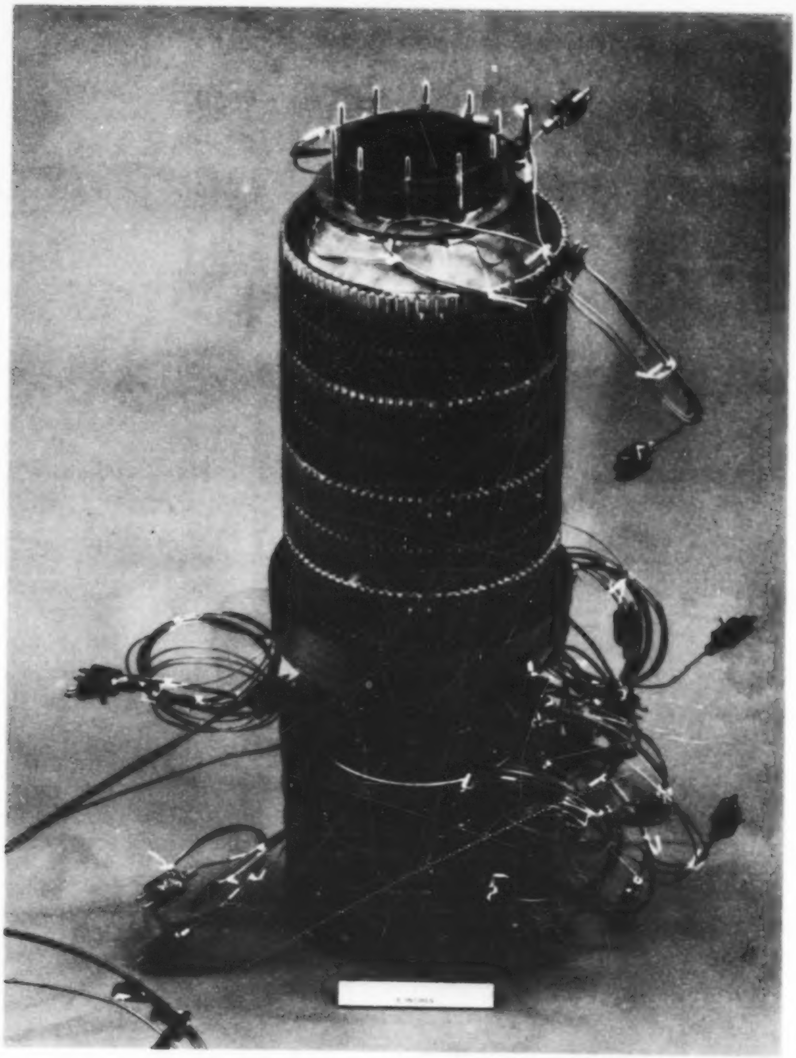


Figure 21. Full-Scale Combustor During Assembly (FRT Version)

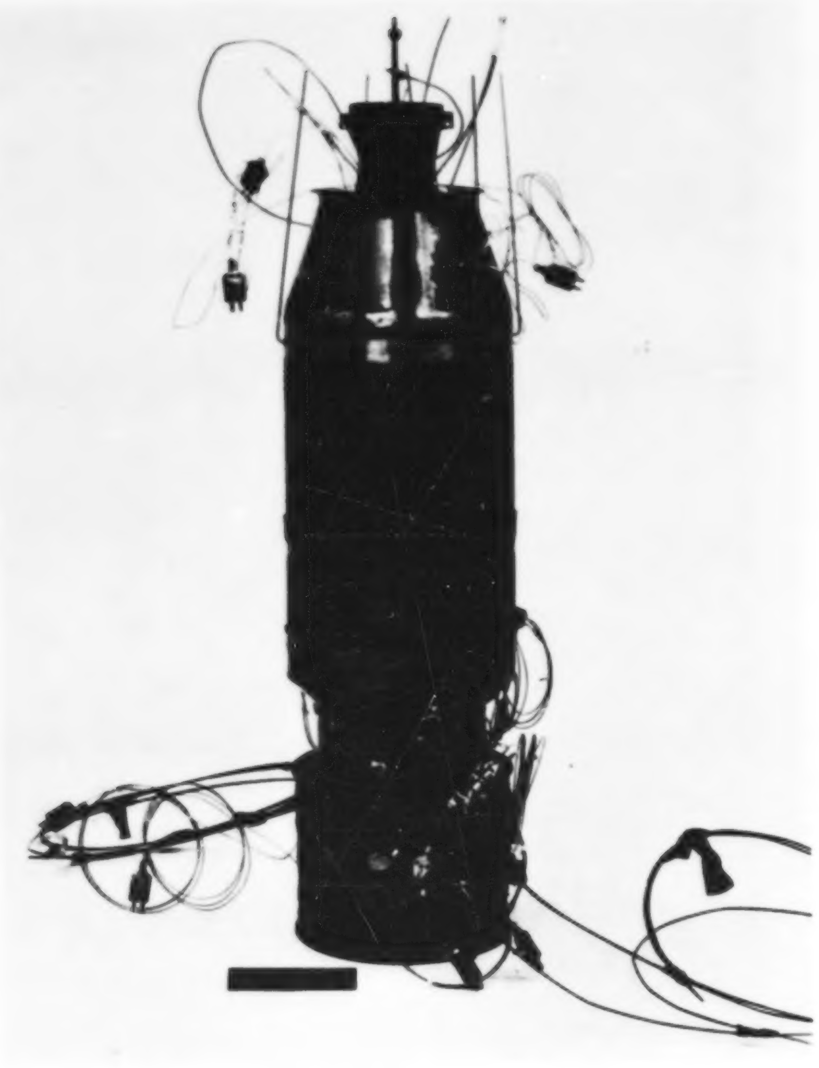


Figure 22. Full-Scale Combustor Fully
Assembled (FRT Version)

TABLE I. DESIGN REQUIREMENTS FOR THE FULL-SCALE PROTOTYPE COMBUSTOR

Type Combustor: can (1 of 8, internally mounted)

Basic Dimensions: 10 inch diameter, 20 inch length

Design Point Requirements:

	(Baseload)	(Idle)
Airflow -	31 lbm/sec	7.8 lbm/sec
Pressure -	188 psia	40 psia
Inlet Temperature -	722°F	285°F
Temp. Rise -	1160°F	625°F

Pressure Drop: 3% combustor, 2.5% diffuser

Lean Blowout: 0.006 fuel-air ratio (burner exit)

Exhaust Emissions (max. at any setting):

	(0% Fuel N)	(0.5% Fuel N)
NO_x	50 ppmv at 15% O_2	100 ppmv at 15% O_2
CO	100 ppmv at 15% O_2	100 ppmv at 15% O_2

TABLE II. AERODYNAMIC MODEL CALCULATIONS FOR FULL-SCALE PROTOTYPE COMBUSTOR

- Configuration for 5.5% Pressure Drop
- Baseload Power Setting (Damper Open)

Station	1	2	3	4	5	6	7	8
Wa(cum) - pps	6.035	6.035	6.035	6.035	6.035	18.105	18.105	29.658
Equiv. Ratio (local)	0.0	1.300	1.300	1.300	1.300	0.433	0.433	0.265
T_t - °F	722	722	722	3686	3686	2505	2505	1875
P_S - psia	187.30	185.72	185.94	185.78	184.97	175.15	177.74	171.71
P_T - psia	188.00	186.94	186.12	185.93	185.93	180.98	178.38	177.66
Velocity - fps	123.4	163.4	61.7	108.7	271.6	578.4	191.1	524.4
Mach No.	0.074	0.098	0.037	0.036	0.090	0.224	0.074	0.228

TABLE III. THE EFFECT OF IMPORTANT PARAMETERS ON DROPLET SIZE

Drop	a	b	c	d	e	f	Reference
$2\bar{r}$	.5	.33	.16	-.16	-.33	-.66	3
$2\bar{r}$	—	.66	.33	-.33	-.66	-1.33	3
$2\bar{r}$	.5	.25	.25	-.25	-.25	-.75	4
MMD	.16	.34	.41	-.84	—	-1.33	5
$2\bar{r}$	—	—	.50	-.5	—	-1.0	6
$2\bar{r}$	.375	.25	.375	-.375	-.25	-.75	7
SMD	—	—	.33	-.37	-.3	-1.0	8
SMD	.375	.25	.375	.25	-.875	-1.0	2
SMD	—	—	—	—	—	-.75	9
$2\bar{r}$	—	—	.33	-.16	-.16	-.66	10
$2\bar{r}$	.166	.333	.50	-.125	-.66	-1.33	11
SMD	.375	.25	.375	-.125	-.5	-1.0	

$$\mathrm{SMD} = K(d_f)^a(\nu_f)^b(\sigma_f)^c(\rho_f)^d(\rho_a)^e(V_a)^f$$

A RESEARCH PLAN TO
STUDY EMISSIONS FROM
SMALL INTERNAL COMBUSTION ENGINES

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ABSTRACT

This paper examines some of the requirements for investigating the environmental status of small internal combustion engines. These engines range in size from 1¼ horsepower to 15 horsepower and power a variety of equipment used by homeowners and industrial members.

With the general growing concern in EPA of identifying sources of potentially carcinogenic emissions, there exists a possibility that these small internal combustion engines are a problem source. Research to characterize emissions from this source has largely been limited to criteria pollutants, even though the small internal combustion engine is an incomplete combustion; therefore some carcinogens and other hazardous compounds are probable.

The basic requirements addressed for an integrated research design include:

a) Analytical Equipment;
b) Experimental System Design; and
c) Statistical Experimental Design.